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Thermal Storage.

DURING the last ten or more years, Mr. Druitt Halpin has endeavoured to convince the engineering world of the benefits to be obtained by his system of thermal storage, as applied to steam boilers. The arrangement is particularly simple, consisting of little more than a cylindrical water-tank placed above and connected to the boiler by means of a pipe. All the feed-water is passed through the cylinder, the water being heated by surplus steam generated during intermittent periods of light-load on the engines, supplied from the boiler. The idea of storing heat in this manner has apparently not proved attractive to engineers generally, possibly because at first view the system does not seem to offer any advantage over methods of feeding boilers with water heated in exhaust steam calorifiers, or economisers. It is a fact, however, that the system possesses distinct advantages, the nature and extent of which can best be explained by reference to the results obtained in two practical tests, one conducted with Cornish boilers and the other with water-tube boilers.

The first of these tests was made in 1894, by Professor Unwin, the boilers being installed at a large hotel in London. One boiler was sufficient for the supply of steam required for the machinery and other apparatus, and the trial extended over two weeks. During the first week one boiler was used in the ordinary way, so that data might be obtained as to the power used and the consumption of fuel and water. During the second week the other boiler, of precisely the same size as the first, was employed in conjunction with the storage-cylinder. Comparison showed that the requirements of the building were practically identical through each week. The results were somewhat surprising, inasmuch as the high evaporative duty of 10 lbs. of water per pound of coal during the first week was considerably improved, the saving in coal being calculated by Professor Unwin at 19·7 per cent.

We will now turn to the second test, which was made this year at the Wood Lane generating-station, belonging to the Kensington and Notting Hill electric-light companies. The reason which led to consideration of the thermal-storage system was, that the engineer, Mr. H. W. Miller, M.Inst.C.E., found it necessary to provide for an additional output of 1,000 kw. and, if possible, without extension of the boiler-house, or of the boiler-plant. On behalf of thermal-storage it was urged that the duty of the existing boilers could certainly be increased without involving the occupation of additional floor-space, and that some saving might be effected as an incidental advantage. The boilers, which were designed to evaporate 12,000 lbs. of water per hour each, are provided with two steam- and water-drums, 24 ft. long

by 4 ft. diameter, and above these are placed the thermal-storage vessels, connected with the boiler-drums by 10 in. stand-pipes, through each of which feed-pipes descend. Each stand-pipe forms a permanent connection to the steam-space of the boiler-drum, and consequently temperature and pressure are always equal in the storage-cylinder and the boiler. All feed-water is pumped into the top of the thermal storage-vessel, where, in passing through the steam, it is heated to boiler-temperature, and enters the feed-pipe through a regulating-cock. During periods of light-load the storage-vessel is filled up, and during periods of heavy-load feed from the pumps is stopped, and the boiler is fed from the water contained in the storage-vessel. Before the installation of the auxiliary-plant it was estimated that the output of each boiler would be increased by 25 per cent. without the slightest risk of priming, but the actual result of trial shows that the possible evaporation has increased by more than 108 per cent., as each boiler is now capable of evaporating 25,000 lbs. of water per hour instead of 12,000 lbs. per hour as before. Naturally, much more coal is consumed, but even in the respect of fuel economy a small saving has been effected.

Referring to the first test mentioned in this article, Professor Unwin stated at the time that he was unable to account, by any recognised theory, for the greatly increased fuel economy, and with regard to the second test it may be said that as the boilers had always been fed with hot water, there seems at first sight no reason why thermal storage should have the effect of permitting the evaporation to be more than doubled. With regard to the last-mentioned feature we may remark that the explanation is probably to be found in the fact that the customary method of boiler feeding is not the most suitable one, for the water is more or less variable as to temperature, and the manner in which it is introduced probably tends to interfere with the regular, but easily disturbed, action which is necessary for the steady formation and gentle liberation of dry steam. In the perfect regularity with which the feed-water descends from the storage vessel we may probably find the true reason for the success of the thermal-storage system. To effect a saving of fuel is an important advantage which should appeal to all owners of steam boilers, but it is now established that the capital cost of land, buildings, and boiler plant can be greatly reduced by the aid of the system to which we refer.

Electricity Legislation.

IT is satisfactory to find that the Government have at last decided to remove some of the disabilities hitherto suffered by electric supply undertakings. The Government Bill introduced into the House of Lords by Lord Wolverton is intended to effect some much needed reforms in the existing Acts relative to electric lighting, and therefore deserves the most particular attention of all who are interested in the subject. It is somewhat unfortunate that the recommendations made by the Joint Committee in 1898 were not acted upon long ago, but some satisfaction must be felt that Parliament has at last been invited to redress some of the wrongs inflicted by misguided legislation in the past. Existing Acts distinctly discourage any company from undertaking electrical works, even in places where the local authority has no desire to inaugurate similar works. The chief reason for this state of things is that the consent of the local authority has to be obtained, and, instead of welcoming proposals for the benefit of the inhabitants, most local bodies seem to be actuated by the wish to stifle and kill enterprise on the part of electrical companies. Even in places where such companies have established supply works, the terms extorted by local authorities are so exorbitant and harassing as to constitute an intolerable burden upon the companies and to involve the payment of higher rates for current than would otherwise be necessary. We are pleased to find in the new Bill a clause depriving local authorities of the power now possessed to veto the grant of Provisional Orders to private companies. Objections may still be raised for consideration by the Board of Trade,

but the absolute veto, with all its concomitant evils, is to become a thing of the past. Another noteworthy reform contemplated by the new Act is a modification of the purchase clause. At the present time, as our readers are aware, the local authorities have the power to purchase the undertaking of an electricity company on most absurdly inequitable terms. Recognising the evil effect of the purchase clause upon electrical enterprise, the Joint Committee in 1898 recommended that this clause should only be inserted in Provisional Orders if the local authorities were able to prove justification for such a course. This recommendation was sensible and just, and it properly laid the onus upon those who coveted the goods of their neighbours. Unhappily, although the Government Bill provides that the Board of Trade may exclude the power of compulsory purchase, it makes the lamentable mistake of protecting the strong, and of placing upon the weak the onus of fighting hard for justice and for the continued possession of their own property. We hope that this reversal of the principle that gives security to the members of a properly organised community will be vigorously opposed before the Bill is finally accepted by Parliament. Other important modifications of existing legislation are to be found in provisions enabling companies authorised to supply electricity to acquire compulsorily the land required for generating stations, whether such land be situated within or without the area of supply, and giving undertakers power to break up any road, railway, or tramway, as may be necessary for the execution of authorised works. Both of these alterations will considerably facilitate the operations of electricity-supply companies, and there are others of less noticeable character which will tend to simplify matters and to conduce to extension of the electrical industry. Unfortunately, the Bill comes too late to ameliorate the lot of electricity companies in the large cities and towns of the kingdom, but this disadvantage might be removed by retrospective treatment.

Terrestrial Heat.

Of radium and radio-active substances the world has heard much during the last year or two. Scientists have communicated to their *confrères* and to the people at large the full extent of their present knowledge, and until some new phenomena are discovered there is little more to be said about these interesting substances. Nevertheless, they still afford food for reflection, and suggest fresh theories for the explanation of difficult points that have heretofore baffled the intelligence of scientific men. A suggestion of this character proved to be the most interesting feature of the lecture recently delivered by Prof. Rutherford at the Royal Institution. It is now known that radium is distributed very widely over the earth, being present everywhere, although in the most minute quantities. No one can say how much heat is provided by the existing supply of radium, nor is it known definitely whether or not the supply of radium is continually being increased by the disintegration or transformation of other substances. Neither can it be asserted definitely that the radium already distributed through the earth is sufficient to account for the gradual increase of temperature which is noticed as distance below the surface is increased. Prof. Rutherford is of the opinion that the amount of radium believed to be present in the earth is quite sufficient to explain the temperature gradient as measured to-day, and to account for all the heat lost from this planet. Geologists and physicists are by no means unanimous in their estimates as to the age of the earth. Geologists base their calculations upon data obtained by observation, and within reasonable limits the results so obtained may be regarded as approximately correct. The computations of physicists have been based upon assumptions and conducted in conformity with established theoretical principles, and until recently it has been difficult to see where any source of error could have originated. Lord Kelvin, however, had the foresight to qualify his calculations by adding the words "provided no new

source of heat is discovered," thereby exemplifying the caution that distinguishes the true Scotsman, and that willingness to learn that should characterise every man of science. In former days we were invited to regard the earth as a slowly cooling mass that would in time become cold and perish by falling into the Sun. We were even taught to measure the heat of the sun, as if it were produced by the combustion of so much coal. Radium has now come, upsetting these ideas, and opening up new vistas of thought, to the end of which the intelligence of man is as yet unable to penetrate.

Gas-Engine Research.

THANKS to the labours of the Gas-Engine Research Committee and of independent investigators, internal-combustion motors are far better understood in the present day than at any previous period. Nevertheless we have much to learn, and many difficult problems still await solution. Difference of opinion still exists upon physical and chemical questions of scientific interest, but it is satisfactory to find that, so far as practice is concerned, engineers, physicists and chemists are fairly in agreement as to certain essential features of gas and kindred motors. These features may be thus enumerated:—(1) initial compression of the working fluid; (2) exposure of the flame to the smallest area of cooling surface for the shortest time; and (3) the use of the lowest flame-temperatures consistent with high available pressures. It is not always the case that the foregoing requirements can be complied with in practice, and those of our readers who desire to inquire into the difficulties, both practical and scientific, which opposed their realisation, will find useful information in the James Forrest Lecture recently delivered by Mr. Dugald Clerk. Notwithstanding the huge strides made in the way of physical knowledge, it is still the fact that the theory of the internal-combustion engine is in a position very much akin to that of the steam-engine before the properties of steam had been accurately determined. Engineers know that an absorption or suppression of heat accompanies gaseous explosions, but they cannot accurately explain either its causes or its proportions under the conditions prevailing in actual practice. The great need of the present day is the definite settlement of data for the design of a quantitatively accurate standard-engine of comparison; and, in an inquiry of this kind, it is necessary that the engineer, the physicist and the chemist should work hand in hand. Before a really reliable engine of this character can be devised, further investigation must be made as to the actual conditions prevailing during gaseous explosions; and one of the first questions to be definitely settled is that of the specific heat of air, and of the various gases entering into the chemical action of combustion, at high temperatures. Le Chatelier and other distinguished scientists have endeavoured to throw light upon such values by means of combustion experiments, but unfortunately there is reason for believing that those investigators were wrong in assuming combustion to be complete at maximum temperatures. Consequently the accuracy of their conclusions may reasonably be doubted. Three methods for determining the specific heat of gases are suggested by Mr. Dugald Clerk, none of these involving combustion. They may be thus briefly stated:—(1) By compressing and expanding the gas in suitable apparatus and after comparing compression and expansion curves together with heat flow by calculating the specific heat. With a pressure of 2,000 lbs. per sq. in. it would be possible to attain a temperature of 1,000 degs. Cent., and the heat added would be readily measured by the mechanical work done. (2) By heating the gas electrically to the required degree, measuring its temperature by the Callender method and absorbing the heat of the gas in a calorimeter. Thus a positive determination of specific heat would be obtained. (3) By causing a gaseous explosion, measuring the inflammable gas not consumed, and estimating the amount of heat evolved at a given temperature. All such experiments are naturally expensive, and to obtain reliable constants would involve years of hard work. The

labour, however, would be amply repaid by the results; and, at a time when the whole world is deeply interested in the perfection of the internal-combustion motor, it is much to be desired that further investigation of the kind here indicated should be undertaken by some organisation with ample facilities for carrying the work to a successful termination.

LONGITUDINAL CIRCULATION IN STEAM BOILERS.

THE problem of increasing the normal circulation in a steam boiler is not a new one, and numerous methods for effecting this have been suggested or introduced, which have met with varying success.

The apparatus about to be described, however, is novel, in that its action produces a flow of water in a longitudinal direction in the boiler. Such a motion is apparently accompanied by several important advantages which will be mentioned below.

The apparatus consists of a series of hoods or cowls placed at both top and bottom of the cross-tubes, and held in position by a stay bolt with a nut at each end. No joint is necessary, and absolutely no alteration to the boiler is needed.

Its action is simple, and need be described in a few words only. The hot gases impinging against the cross-tubes set up a rapid evaporation of the water, the steam rising and seeking the nearest outlet. This is the inclined mouth of the top hoods, from which the steam and hot water are discharged in a longitudinal direction with an intermittent action, the steam escaping into the steam space, and the water travelling along the top of the furnace flue by the momentum set up. As the water is evaporated out of the tubes, other water comes in to take its place, which it can only do by entering in at the mouth of the bottom hoods.

A flow of water is thus set up in a direction starting from the front of the boiler at the bottom, in through the inclined mouths of the bottom hoods, up the cross tubes, out through the mouths of the top hoods, and along the flue and furnace-tops to the front of the boiler, sweeping the tops clear of the blanket of steam, and gradually falling to the bottom of the front end of the boiler, to be again drawn in and swept out as before.

The importance of such an action of the water, as it is claimed this apparatus sets up, cannot be exaggerated; and in raising steam on a cold boiler, working pressure has, we are informed, been obtained in little more than half the time usually taken, and with perfect safety, the difference in temperature which ordinarily exists between the top and bottom of the boiler being much reduced.

In the various tests of the circulator that have been carried out, it has been found that the efficiency varies in proportion to the number and position of the cross-tubes, and that the nearer the latter are to the bridge the greater the saving in fuel. This is due to the fact that it is on that part of the furnace-crown, from a little in front to a little behind the line of the bridge, that nearly 75 per cent. of the evaporation takes place. The steam-bubbles are here formed much more rapidly, and the blanket of steam is heavier, than in any other part of the flue. It would appear, then, that the better the circulation of the water over this space the faster the steam-bubbles are washed off, and consequently the more steam is generated.

A series of tests extending over a period of several weeks were carried out at the Northampton Institute under the direction of Mr. Chas. E. Larard, A.M.Inst.C.E., Chief of the Mechanical Engineering Dept., and the results of which have been furnished by the inventor.

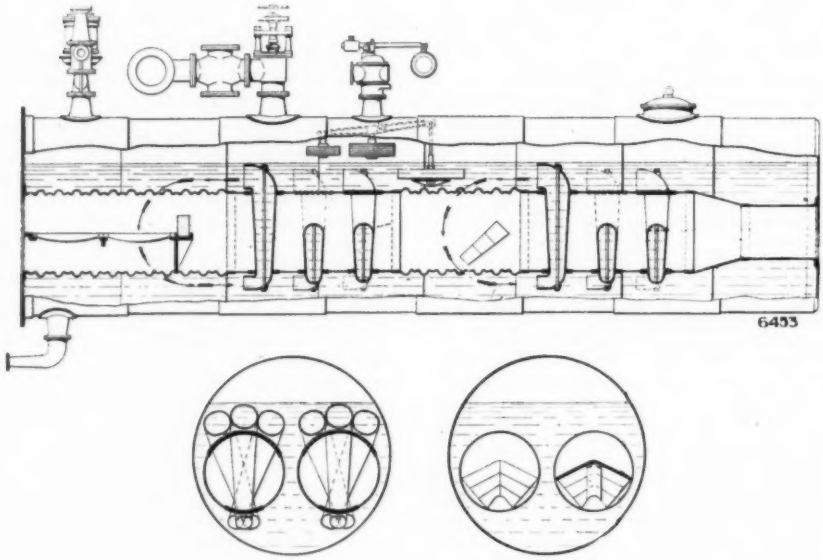
The boiler under test was an Adamson boiler, 20 ft. long, with two cross tubes in each furnace and a rated output of 3,000 lbs. of steam per hour, and fitted with Meldrum furnaces. The heat value of the coal used, as determined by a calorimeter, was 12,240 B.T.U. and the percentage moisture 2½ per cent. The working steam pressure for both trials was kept as near as possible to 150 lbs. per sq. in. The dampers were fitted with graduated quadrants, so that any degree of damper opening could be obtained to regulate the draught, and graduated arcs were also

supplied to the steam cocks on the Meldrum blast pipes, so that in the trials, with and without the circulators, identical conditions of working were aimed at, both with respect to draught, blast pressure, and quantity of steam from blast.

Throughout the trials the water-level was maintained as nearly as possible to a pre-arranged mark, the boiler feed-pumps supplying the water from graduated feed-tanks at the rate of about 3,000 lbs. per hour. A very good check on the water-supply was also obtained from a counter fixed to the feed-pump which registered the strokes throughout the long trials.

upper part of the boiler were taken as corresponding to the temperatures of the steam at the particular pressures at the times when the observations were taken.

At the commencement of the trial, without the circulators in use, steam was raised from clear grates (banked fires being pulled out) at 21 lbs. boiler-pressure. At the time the fires were lighted the temperature difference between the top and the bottom was 15 degs. Fahr.; 30 minutes later, steam pressure being 65 lbs., the temperature difference was 44 degs. Fahr.; while 45 minutes later the temperature difference was 48 degs. Fahr., with a boiler pressure



The coal used was first weighed out in lots of about 200 lbs., and then tilted on to the stokehold-floor ready for use, the times of commencing and finishing each lot being carefully noted.

The two long trials were each of seven hours' duration, after full steam was raised.

In order to obtain some information about the temperature difference between the top and bottom of the boiler, a hole was drilled through the mud-hole cover at the bottom, and a thermometer in a suitable case inserted, and the temperature taken from time to time. The temperatures of the

of 75 lbs. The maximum temperature-difference was 71 degs. Fahr., and reached in an hour from the commencement of the trial to the full boiler-pressure. From the time of raising the full-steam pressure to the end of the seven-hours' trial, the temperature difference steadily fell until a difference of 38 degs. was obtained. The corresponding results from the boiler with the circulators in use were 27 degs. Fahr. at the end of 30 minutes, 43 degs. Fahr. in 45 minutes, 52 degs. Fahr. maximum at the end of an hour, and 32 degs. Fahr. at the end of the trial.

Results of 7 hours' Trial (two cross-tubes to each furnace).

	Without Circulator.	With Circulator.
	Lbs.	Lbs.
Rate of coal consumption per hour	442	416
Feed-water delivered to boiler ,,	2,997	3,014
Temperature of feed-water (° Fahr.)	53	52
Evaporation per lb. of coal as fired	6·62	7·08
Equivalent from and at 212° Fahr.	8·04	8·59
Evaporation per lb. of dried fuel ...	6·79	7·26
Equivalent from and at 212° Fahr.	8·24	8·82
Thermal efficiency of boiler ..	65·49	70·08
Efficiency gained	—	7%

A further series of tests was taken at the Borough of Islington Electricity Works on two Lancashire boilers, each 30 ft. by 7 ft. 6 in., built by Daniel Adamson, with four cross-tubes in each flue, the first tube being only 2 ft. behind the bridge. Seven pairs of hoods were fitted, and six tests in all were taken, each of 9 hours' duration. The fuel was Crynant coal, and Meldrum's forced draught was used on each test.

The principal figures are appended :—

	Without Circulator.	With Circulator.
Total water evaporated..	4,685 gals.	4,792 gals.
Average feed temperature (° Fahr.)	115°	113°
Average steam pressure..	132 lbs.	133 lbs.
Total coal consumed ...	45 cwt.	42 cwt.
Water evaporated per lb. of coal	9·28 lbs.	10·18 lbs.
Equivalent from and at 212° Fahr.	10·63 lbs.	11·7 lbs.
Efficiency gained ...	—	10·6 %

The most recent tests were taken at the Leyton Public Baths, where comparative figures for eight hours were obtained, with and without the Circulator.

The boiler under test in this case was a 30 ft. by 7 ft. Lancashire, with four cross-tubes in each flue, all of which were fitted with the hoods. The steam was used for supplying an engine driving a deep-well pump, which had to be run at a very slow speed in order not to pump the water out of the well faster than it ran in, which accounts for the low evaporation per hour. The atmospheric conditions for each test were practically identical and need not be taken into account. The coal used was Welsh (Fforchwen).

	Without Circulator.	With Circulator.
Coal consumed per hr. ..	210 lbs.	168 lbs.
Water evaporated per hr.	1,976¼ lbs.	1,862½ lbs.
Total coal consumed ...	15 cwt.	12 cwt.
Total water evaporated .	1,581 gals.	1,490 gals.
Average steam pressure. (° Fahr.)	80°	80°
Average feed temperature. (° Fahr.) ...	175°	176°
Water evaporated per lb. of coal	9·4 lbs.	11·08 lbs.
Efficiency gained ...	.....	15·1 %

The Schofield Patent Circulator can be applied to almost any type of shell boiler, and the sole proprietors are Circulators, Ltd., 147, Leadenhall Street, London, E.C.

We understand that a series of trials on marine boilers is to be made shortly, the results of which we await with interest.

BOOK REVIEWS.

Entropy; or Thermodynamics from an Engineer's Standpoint, and the Reversibility of Thermodynamics. By James Swinburne. 1924. (Westminster : Archibald Constable & Co., Ltd. 4*s.* 6*d.* net.)

In this volume we have presented to us in a convenient form Mr. Swinburne's views on entropy, and his criticisms of the treatment of that subject in modern textbooks. Readers of THE ENGINEERING REVIEW will scarcely need to be reminded that this contribution to our technical literature had its origin in

the fierce controversy which followed the delivery of Mr. Swinburne's presidential address to the Institution of Electrical Engineers, and that it first appeared as a series of articles in *Engineering*. The author's treatment of his subject, therefore, does not come before us now with the freshness of something quite new, and as we have already given, in special articles and in our abstracts of engineering literature, the substance of his writing, it will not be necessary to repeat it here. But we welcome the present volume, and are glad that Mr. Swinburne has published his views on entropy in book form, as we consider that a real service has been done by him to engineers and students generally by his protest against the current text-book method of treatment, and by his exposition of the shortcomings of that method. We are aware that some professors may, and do, consider Mr. Swinburne's views to be extremely dangerous ; one of them has said so in the columns of a contemporary in language which is less than polite, and far from creditable to himself ; but the fact remains that competent engineers and scientists have followed with keen interest Mr. Swinburne's contest with the professors ; and, if one is to judge by writings in the technical press generally, the verdict is not for the professors. Besides the engineering articles the volume contains, as an appendix, Mr. Swinburne's paper to the Mathematics and Physics section of the British Association last year. This is entitled "The Reversibility of Thermodynamics," and gives a good summary of the ordinary text-book treatment, and of Mr. Swinburne's proposed treatment of thermodynamics. It would be easy to single out statements here and there in the volume to attack as inaccurate or inexplicit, and Mr. Swinburne's critics have not hesitated to make the most of these opportunities ; but these are minor matters and do not seriously affect the real value of Mr. Swinburne's work, which is, that he has shown more clearly than has been done hitherto the physical importance of entropy, and the errors and limitations to which the confusion of reversible with irreversible changes leads. We recommend students of engineering to read this volume.—*W. C. H.*

Steam Boilers : their Theory and Design.—By H. de B. Parsons, B.S., M.E. (Longmans, Green & Co. New York and London. 375 pp., 160 illustrations. 10*s.* 6*d.* net.)

This book is by an American author, a consulting engineer and a professor in the States, and hence not unnaturally deals more particularly with American boiler-plant than an English author would consider desirable.

The author first of all gives a glance at some elementary physical definitions, most of which might very well be omitted. The reader of a book like this does not require to be told what "solids" and "fluids" are, nor to have the most elementary conceptions of heat explained. This kind of padding for books is becoming very common, but it would be much better if such things were replaced by a concise table of physical quantities in the form of an appendix.

Having treated clearly of combustion and fuels—not omitting liquid fuels—the author passes on to consider briefly furnace temperatures (which he under-estimates) and boiler efficiency. In our opinion he would have done well at this stage to have treated of what is sometimes called the science of steam production. As it is, the book is rather too empirical, not that we decry the information obtained by experience, but if progress is to be made we must use our experience as the basis of a working theory. A careful analysis of some of the well-known boiler tests, together with a clear explanation as to how the heat gets from the hot gases to the water would add much to the value of the book.

We are glad to see that the author treats pretty fully of chimneys and draught, both natural and artificial. Chimneys are too important to be left to the tender mercies of the local bricklayer. Mechanical stokers are briefly considered, each of the three types illustrated being, however, almost unknown in this country. There is also a good chapter on smoke prevention. The treatment of impure feed-water is considered, but the author seems to have forgotten the water-softener for purifying the feed before it enters the boiler.

Superheated steam and superheaters are treated briefly in an appendix. Mention of superheated steam reminds us that the author, when mentioning the use of copper steam-pipes, has forgotten to point out that they deteriorate when used with superheated steam. In connection with steam pipes some mention of the seriousness of water-hammer would not be out of place, and a detailed drawing of a well-designed steam range would have been very useful as illustrating the points to be kept in mind with regard to steam piping.

In spite of its defects the book contains a good deal of useful information, and is interesting as illustrating boiler practice in the States. The book is well got-up and illustrated.—*F. F.*

The Theory of Steam Turbines.—III.

By FRANK FOSTER, B.Sc.

(*Continued from Vol. 10, p. 465.*)

Buckets: Riedler-Stumpf Turbine.

In principle the Riedler-Stumpf turbine is very like the Curtis. The steam is fully expanded in the steam nozzle (Fig. 3, Art. I) and delivered into the buckets (Fig. 16). It leaves the buckets and enters the guide-passage, which takes it across the rim and delivers it into another bucket or a fresh set of buckets. The nozzle and guide entrance and exit are in planes perpendicular to the shaft.

Referring to Fig. 17, AB is the velocity of the steam leaving the nozzle, and CB that of the buckets. Then θ is the angle of the nozzle to the rim, and α that of the buckets to the rim. AC is the velocity of the steam relative to the wheel at the inlet, and CD the relative velocity at exit, CE being the absolute velocity, and ϕ the angle of the guide to the rim where the steam enters the guide. Allowances for friction losses should be made, as for the Curtis turbine. There will be probably rather a large loss in the long guide-passages. The calculation of the angle of delivery of the guide to the rim proceeds exactly as if the guide were a nozzle, the angles of the buckets not necessarily being the same as in the first case, if, as is usual, there are two sets of buckets. If there is only one set of buckets on the rim, this must be remembered, since it fixes the angle of delivery. Care is necessary in fixing the circumferential positions of the guides relative to the nozzles, so that the steam leaving the wheel actually enters the guide. This is especially necessary in small-powered turbines where there are only a few nozzles and guides, so that a small error in the position of the guide will cause the steam leaving the wheel to miss it entirely. Since the advance of the guide in front of the nozzle depends on the ratio of the velocities of the steam and the wheel, it follows that, unless provision is made for rotating the guides slightly on the shaft, the velocity of the steam leaving the nozzle must be reduced proportionally to any reduction of the wheel-speed, such as occurs in naval work, where the speed of rotation depends on the power exerted. Hence throttle-governing must be employed where the speed varies. The same remarks apply, although not so forcibly, to a Curtis turbine, and in a less degree to a Parsons turbine—indeed, it is quite possible that, in a Parsons turbine, especially a large one, that the loss which occurs in throttling would more than balance any gain in the turbine proper.

Governing.

There are three methods of governing constant-speed turbines. The simplest is that of simply throttling the steam supply. This is wasteful, as, whilst reducing the quantity of steam admitted, it reduces its available energy. Thus, if steam at 135 lbs. absolute, containing 2 per cent. of moisture, be throttled to 65 lbs. absolute, its available energy drops from 304 to 264 thermal units, a loss of 13 per cent.; back pressure, 1 lb. absolute.

A method used frequently on De Laval and Curtis turbines is to govern by varying

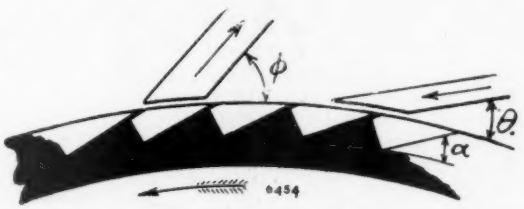


FIG. 16.

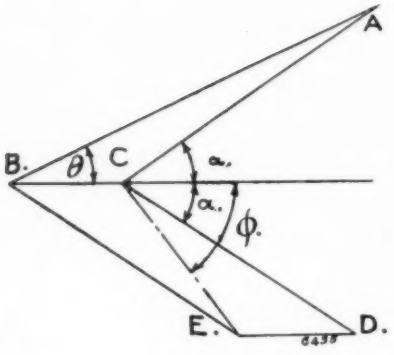


FIG. 17.

the number of nozzles in action, with recourse to throttling only for intermediate powers. This method reduces the number of streams of steam passing through the turbine. This is a source of loss, but probably less so than throttling.

Unless judgment is used in selecting the nozzles to be put out of action, the active nozzles will be unsymmetrically spaced, and thus introduce side-thrust on the bearings—probably not serious, however, as the main pressure on turbine bearings is that due to the weight of the rotor.

The method of governing in a Parsons turbine is as follows: The turbine-shaft has a worm on it, driving a worm-wheel. On the worm-wheel shaft—which also drives the oil-pump—is a cam which operates a lever controlling the valve of a steam relay. This steam relay controls the double-beat admission-valve, and hence this is opened once per revolution of the worm-wheel—about once in 30 revolutions of the turbine-shaft—admitting a blast of full-pressure steam into the turbine. The duration of this blast is controlled either by a centrifugal ball-type of governor or by an electrical solenoid fitted to the dynamo. The solenoid is attached to one end of a lever, the fulcrum of which is moved by the cam mentioned. The short end of the lever is connected to the steam relay. Fig. 18 shows the variation of pressure just inside the turbine casing at about half-load.

Initial Condensation.

This method of governing introduces initial condensation, especially at light loads. As will be seen, the pressure and temperature fall greatly between the blasts, and thus cool the wheels and casing, so that, when the hot incoming-steam of the next blast enters, some of it is condensed. The use of pretty highly superheated steam will reduce this condensation and lead to economy. It is sometimes said that this method of governing is equivalent to "expansion" governing on reciprocating engines. This is not so: whilst the blast is on we may say that the governor is producing no effect on the steam distribution. Between the blasts the pressure at the supply-end falls off rapidly, but in the same manner as if throttled; so that really the results will be similar in a general way to those obtained by using throttle-governors. This is borne out by the straight-line curve representing the relation between total feed and power generated.

Vibrations.

This method of governing by blasts may also set up vibrations, more especially at light loads. The motion of the steam in the turbine, besides rotating the wheels, also produces an end-thrust and a turning moment on the casing, due to the reaction of the steam on the vanes of the guides. This force on the casing will vary with the velocity of the steam—that is, every time the pressure falls—so that the turbine casing will be subjected to a

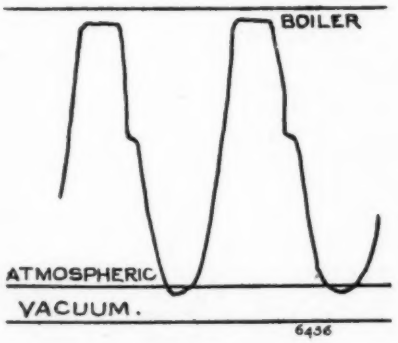


FIG. 18.

periodic force which will set up vibrations, small in general but capable of becoming serious under circumstances favourable to them.

It is not practicable in a Parsons turbine to govern by stopping one of several streams of steam as in the Curtis turbine, not because of there being only one admission-valve —since it would be easy to have several, each admitting to a portion only of the first guide-ring — but because in this turbine the pressure varies from wheel to guides and guides to wheel along the shaft, so that there must be a continuous flow through all the guide openings. We might as well imagine, if we had two tanks of water connected by, say, six pipes and the higher tank fed by six pipes, that by closing, say, three of the admission pipes the water flowing into the lower tank would confine itself to three pipes only. In the Curtis turbine there is no difference of pressure to speak of from wheel to guides and guides to wheel, so that the steam will not move from one to the other, unless impelled by steam forced out of a nozzle.

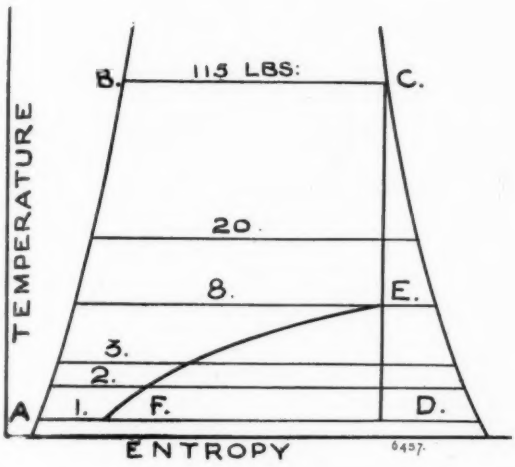


FIG. 19.

Condenser Pressure.

It is well known that a turbine is able to make better use of a good vacuum in the condenser than a reciprocating engine. The reason is twofold—in the first place there is usually a greater drop in pressure between the cylinder and condenser of a reciprocator than in a turbine; and, secondly, the reciprocator has to release the steam at a comparatively high pressure, and cannot therefore take full advantage of the greatly increased volume of the steam at low pressure. This is illustrated in Fig. 19. The limits of pressure *B C* and *A D* are 115 and 1 lb. absolute. The maximum work obtainable from the steam turbine is represented by the area *ABCD* and is 233,000 foot-pound-second units. The work obtainable from the reciprocator, assuming it to release at 8 lbs. absolute. is only 193,000 units (represented by the area *ABCEF*) or 21 per cent. less than the turbine. The lower the stop-valve pressure the more marked is this difference. Thus, if we reduce the stop-valve pressure to 20 lbs. absolute—other pressures unaltered—the works obtainable from the turbine and reciprocator are 129,000 and 62,000 units respectively, so that the turbine is more than twice as efficient as the reciprocator. In practice the difference would not be quite so marked, but it would still be very great. It is, however, clear that the high economy obtainable from a turbine is largely due to its being able to make good use of low-pressure steam. Down to release in the low-pressure cylinder, the reciprocator is more efficient than the turbine. Thus, according to a curve of water consumption for a large Curtis turbine, as given by Mr. Emmet, the steam consumption running non-condensing is double that running condensing on a 27-in. vacuum. Similar, though rather less marked results, are obtainable with a Parsons turbine. Hence, to give the highest steam-economy, a non-condensing reciprocator exhausting into a condensing turbine with a high vacuum should be used. This is by no means

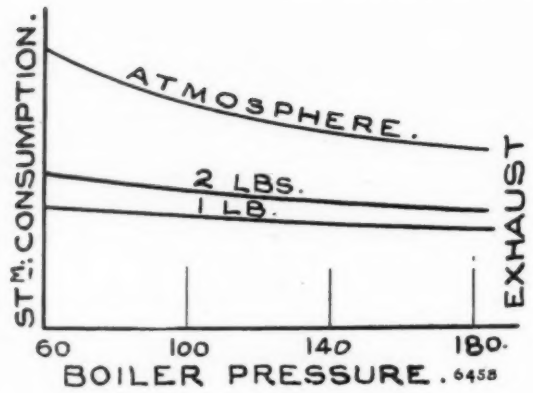


FIG. 20.—SHOWING ECONOMY AT DIFFERENT EXHAUST PRESSURES.

always a possible or desirable combination, but it is so in many large electrical stations and on board ship.

Marine Turbines.

There are two main objections to the use of turbines on ships. Firstly, there is the lack of manœuvring power, and secondly, the low efficiency of propellers running at the high speeds necessary when using turbines. For naval work, where most of the cruising is done at a reduced speed, there is the further objection that the turbines are very uneconomical when running at a speed for which they are not intended. Table IV. is given by Mr. Parsons as showing the relation of the feed-water per effective propulsive horse-power to the boat speed. The low economy at reduced speeds is very marked.

TABLE IV.

Knots	10	12	15	20	26	30
Feed per h.p.-hour ...	90	80	56	44	36	31

On large ships we might use two sets of reciprocating engines, exhausting at about atmospheric pressure into a turbine on a central propeller-shaft. About two-thirds of the power would be obtained from the reciprocators, which would run at their normal speed, and be capable of reversing with perfect ease; the act of reversing cutting out the turbine and exhausting direct into the condenser. These engines would run at their normal speed, and thus obtain the best propeller efficiency, as well as steam efficiency. The central condensing-turbine would be so much more efficient than the low-pressure cylinders of the reciprocators would have been as to more than balance the inefficiency of its propeller. The writer estimates the saving in coal at about 15 per cent., or an equivalent increase in speed. Such an arrangement would be more efficient and quite as handy as any of the combinations of reciprocators and turbines or turbines alone at present in use.

Marine turbines differ from turbines for driving dynamos mainly in having a much lower speed of rotation, and in being split up into separate cylinders or turbines.

The Settling of a Turbine-wheel.

When a weight rotates it always happens, no matter how well balanced it may be, that the centre of gravity of the weight receives some slight—imperceptible by ordinary means of measurement it may be—deflection from the axis of rotation. When so deflected the rotation of the mass about the axis sets up a centrifugal force tending to increase the deflection.

This centrifugal force always acts radially from the axis through the centre of gravity, but its action is simpler to conceive if we resolve it into two component forces acting at right angles to one another, and in proper relative phase. Each force will be simple harmonic, tending to bend the shaft to and fro along the line of action of the force. The combined effect of these two forces is to produce a rotary motion of the shaft (with its attached weight) about the true axis of rotation. These bendings or deflections become serious when the forces causing them have the same frequency as the natural

period of vibration of the shaft and its attached weights; for at this speed the two sets of vibrations fall into step; and, just as a very slight but well-timed force applied periodically to a pendulum will give to it a very big swing, so this small centrifugal force will rapidly increase the deflection of the shaft, and with it the centrifugal force, until the shaft breaks. Hence, in order to determine this "critical speed," all that we have to do is to calculate the frequency of the shaft and attachments. A plain shaft, or a shaft with two or more wheels on it, has more than one critical speed, depending on the modes of vibration of the shaft; but if there is one wheel only, there is only one mode of vibration and only one critical speed

In order to calculate this critical speed:

If W is the weight of the rotating mass and a portion of the shaft (the deflection of the shaft is not the same at all points along it, and hence only a portion of its weight can be considered as dynamically situated with the rotating mass).

F the force (in pounds weight) necessary to bend the shaft 1 ft. at the point of attachment of the wheel.

a and b the distances in feet of the wheel from the bearings.

Then, the revolutions per minute at the critical speed are

$$N = \frac{60}{2\pi}\sqrt{\frac{Fg}{W}} = 9{\cdot}55\sqrt{\frac{Fg}{W}}$$

for a round shaft of diameter d ft.

E is the modulus of direct elasticity in pounds per sq. ft., and

I the moment of inertia of the cross-section $= {\cdot}0482\ d^4$.

Then, if the bearings do not fix the direction of the shaft there — and few bearings do—

$$F = \frac{3EI(a+b)}{a^2b^2}$$

Substituting this in the preceding formula, we can easily calculate N the critical speed. If the bearings fix the direction of the shaft there, then there is no simple general formula for calculating F, but approximately the critical speed will be 2·27 times that calculated from the above formula.

For example, a turbine wheel on a $1\frac{5}{16}$-in. shaft weighs 200 lbs. and is 2 ft. and 1 ft. from its bearings. Then, taking half the weight of the shaft as existing at the wheel, the critical speed is 980 revolutions per minute.

Professor Dunkerly has shown that if N_1 and N_2 are the separate "whirling" or critical speeds of a shaft and a wheel on it, that the resultant whirling or critical speed is

$$N = \frac{N_1\ N_2}{\sqrt{(N_1^2 + N_2^2)}}$$

Applying this to the previous example, we find $N_1 = 4{,}860$ (shaft) and $N_2 = 1{,}000$ (wheel), hence $N = 980$.

This critical speed is usually about one-sixth or one-eighth the working speed for a De Laval turbine.

Stresses in Turbine Wheels.

Owing to the high speed necessary in the De Laval and some other turbines, great care is necessary in designing the wheels. The stresses in a flat disc without any hole or only a small hole at the centre, are (when rotating) greatest at the centre, and hence if a smash occurs it will begin at the centre, and the whole disc will go to pieces. This is most undesirable: it is much better if only a few vanes fly off, and hence the greatest stress should be at the rim. This is attained in the ordinary open-armed wheel, but the maximum working-speed is low. Thus for cast-iron with a stress of 3 tons per sq. in. a rim speed of about 260 ft. per sec. only is possible. For cast-steel having a stress of 9 tons per sq. in. a speed of about 430 ft. per sec. may be attained. Owing to the centrifugal loading of the vanes these speeds must be reduced or the stresses will be higher. In the De Laval and some of the Riedler-Stumpf turbines these speeds are not high enough. To enable higher speeds to be attained wheels that are either solid throughout, or solid except for the shaft-hole, are used. By thickening these wheels at the centre, the metal at the centre running at a low linear speed, relieves the outer portion of the wheel of some of its centrifugal loading.

By suitably adjusting the thickness of

the metal at different radii almost any distribution of stress can be obtained. Two main objects must be kept in view. Firstly, at no place should the stress exceed a certain maximum (to within say 10 per cent.) depending upon the material; and, secondly, the maximum stress should occur at the rim. How to design a high-speed wheel to have a given distribution of stress was pointed out in the *Engineer*,* account being taken of the centrifugal loading of the vanes on the rim. The solution there given is quite sufficiently accurate for all practical purposes, what error there is being between the rim and the centre, where it is of little consequence. Solid wheels bolted to flanges on the shaft should be used except for very small wheels, which should be forged solid with the shaft. Wheels with a hole in the centre for the shaft are not to be recommended. The results of the investigation are very simple, and there is no difficulty in applying them; but as they are rather complicated to write the reader is referred for details to the article in *The Engineer*.

End-Thrust in a Parsons Turbine.

We made the angle of entrance to the vanes a right angle and the angle at exit we called θ (see Fig. 13, Article II.), whilst v_1 was the velocity of the steam relative to the vane. Then the reaction of each pound of steam on the vane perpendicular to the direction of the exit is $\frac{v_1}{g}$ lbs., and the component along the shaft or end-thrust is $\frac{v_1}{g}\cos\theta = \frac{V}{g}$. Hence the total end-thrust for any one stage in which the wheel velocities are all alike is $\frac{V}{g}$ multiplied by the pounds of steam per second multiplied by the number of wheels. Adding the end-thrusts for each stage the total end-thrust is obtained. This end-thrust is balanced by means of pistons rotating with the shaft, on the circumferences of which grooves are cut, into which brass rings, attached to the casing, fit and prevent the leakage of steam past the pistons. There are usually three such pistons in the turbine, each balancing the end-thrust of a section of the wheels. Splitting up the balancing of the end-thrust among three pistons placed one behind the other reduces the tendency to leakage past them; it also gives a better balance when the stop-valve pressure varies —as it does periodically—and the velocities of the steam in the different sections of the turbine do not vary proportionally. The skin friction of the steam passing through the vanes will increase the end-thrust somewhat beyond that calculated as above.

In the example previously worked out there were in that one stage 10 wheels, the velocity of the wheel being 300 ft. per sec., and the steam used per second was 11·5 lbs. The end-thrust for this stage is 1,080 lbs. Taking the pressures on the two sides of the balance-piston at 135 lbs. and 59 lbs. absolute, the area of the piston should be 14·2 sq. ins. The Parsons method of governing by making the time that the pressure is acting on the pistons practically the same as the time that the steam is flowing, makes the balance nearly equally good at all loads. In the above calculation no allowance was made for the friction of the steam on the vanes; this would probably increase the end-thrust by from 5 to 10 per cent. The balancing might be done by ball-thrust bearings, especially in the smaller turbines, this method being both simpler and lighter than that usually adopted.

In the Parsons turbine there is a small thrust-block on the shaft to take up any end-thrust not balanced by the pistons, and also to adjust the clearance between the wheel and guide vanes, the top half of the thrust-bearing being adjustable parallel to the shaft for this purpose.

In marine work the end-thrust of the turbine can be made to partly balance the thrust of the propeller. Thus, if

W = work done by turbine, foot-pounds,
kW = work actually done by propeller,
v = speed of ship in feet per second,
V = wheel speed in feet per second,
n = number of wheels in turbine,
F = lbs. of steam used per second,
P = thrust due to propeller,
T = thrust due to turbine,

* *The Engineer*, Jan. 8, 1904: "The Design of Steam Turbine Discs," by Frank Foster, B.Sc.

$$P = \frac{kW}{v}$$

$$T = \frac{V}{g} \times n \times F$$

$$= \frac{V}{g} \times \frac{gW}{V^2}$$

$$= \frac{W}{V}$$

Hence the turbine end-thrust balances the propeller thrust, if

$$\frac{kW}{v} = \frac{W}{V}$$

$$v = kV$$

if $k = \cdot 7$ and $v = 35 \cdot 5$ (21 knots), then V is only 50·7, which is much too small.

It is not proposed to enter here very fully into the details of turbine construction, but there are a few points of interest.

Vanes.

The vanes of a Parsons turbine are usually made of brass rolled in lengths of the correct section. The individual vanes are simply cut off these rolled bars to the required lengths. A shallow groove about a $\frac{1}{4}$ in. deep and of the same width as the vanes, is turned in the circumference of the rotor or casing. The vanes are placed radially in the grooves, and the space left by the groove between each consecutive pair of vanes is filled by a brass caulking-piece. The De Laval vanes have an enlarged end fitting a corresponding groove on the rim, being thus dovetailed and wedged into position. Many other methods of fixing the vanes have been tried. Rings shrunk on to the outside of the vanes have been tried, and also electric welding. The latter method makes a good strong job, but leaves the depth of the vane passage a trifle uncertain, and, above all, makes it well-nigh impossible to execute renewals in many situations. Soldering and cutting from the solid rim have also been tried. This last is open to the same objection as the welding process.

As to the material of the vanes, brass, copper, and nickel steel have been mostly used. Anything easily oxidisable, like steel, should not be used, as rusty vanes cause a great deal of friction. Copper is said to stand highly superheated steam well, which is contrary to experience in other directions, but may be accounted for by the fact that, owing to the initial condensation in a Parsons turbine, the actual temperature of the vanes is never so high as that of the steam.

It is very important that the passage areas should be correct. This is obtained partly by blanking up some of the guide passages, but also by altering the depth of the vanes. This is most easily attained by boring the casing a true cylinder, and turning the rotor everywhere to the correct diameter.

The vanes usually vary from about $\frac{1}{2}$ to $\frac{3}{4}$ in. in axial breadth, and the pitch rather more than half this.

The clearance possible between the vanes on the rotor and those on the casing is limited by two things. Firstly, there is the question of accuracy of workmanship, the effect of which is too obvious to need comment. The other cause limiting the clearance is the fact that the rotor and the casing expand unequally, being at different temperatures when running but at the same temperature when standing. The thrust block fixes the relative position of the rotor and casing at one end. The other end must be left free to expand slightly, or the casing will hog, and the circumferential clearance will have to be increased if the vanes are not to strip. Thus, assuming the materials used to be cast-iron and the casing to be 10 ft. long, a difference of mean temperature between the casing and the rotor of 70 degs. Fahr. will cause a relative movement of a twentieth of an inch. Hence, we must thoroughly lag or jacket the casing, and we must keep it as short as possible. This latter is obtained by using narrow vanes and high speed (small number of wheels). Splitting the turbine up into separate cylinders reduces the relative expansion per cylinder; but if the cylinders are all on one shaft it aggravates the trouble by increasing the total length, and by anchoring the separate cylinders (at one end only) to the cold bed-plate. Obviously, the clearance must be greater when using superheated steam than when using saturated steam. Wear on the bearings must be practically prevented, or taken up as it

occurs, as otherwise the rotor will sink until it touches the bottom of the cylinder, and blades will be stripped off.

Bearings.

Except for the very highest speeds, the bearings are ordinary bearings, lined with white metal, and of from 4 to 7 diameters in length. The size of a bearing is mainly determined by the heat generated in it, which is proportional to the intensity of pressure, the total area, and the journal speed. Although normally well supplied with oil, the supply may be temporarily cut off, so that a limited supply of oil is assumed in calculating the bearing area, and under these conditions the heat should be radiated as fast as it is generated without undue rise of temperature. This means that the product of the surface speed and the pressure per square inch of bearing surface (which determines approximately the thickness of the oil film, and hence the heat generated) should be approximately constant. This constant varies in practice from about 60,000 to 150,000 where pressures are in pounds per square inch, and the speed is in feet per minute. In one instance of a large Parsons turbine cited by Mr. Chilton, the value of the constant was 90,000. The surface journal speed being high, the pressure must be low, so that a long bearing is necessary. Practically, the only pressure on the bearings is that due to the weight of the rotor.

Steam Jackets.

Except to keep the casing warm and thus reduce the clearance between the vanes, steam-jackets are of no use with highly superheated steam; but otherwise, owing to the initial condensation in a Parsons turbine, jackets would be of use at partial loads provided the turbine steam itself is taken through the jackets and passed through a separator before use in the turbines. Jackets are unnecessary on any of the other types of turbines at present in use, since in them there is no fluctuation of temperature to speak of in the cylinder.

Gas Turbines.

The great trouble with these lies in the exceedingly high temperatures to be handled. In any gas turbine in which the gas actually goes through the turbine —there are some in which the gas propels water through a hydraulic turbine—the De Laval principle must be adopted, so as to reduce the temperature as far as possible before entering the turbine. No great advance in economy beyond that attained by the best reciprocating engines can be expected, as already about 60 per cent. of theoretical work is obtained.

Turbo-Air Compressors.

These really consist of a reversed steam turbine, using, however, air instead of steam. The method of design is the same as for a turbine.

Research.

There is a wide field for research in connection with steam turbines. Mere tests of steam consumption at different loads and with different degrees of superheat tell us very little. What is wanted is systematic tests on portions of turbines under perfectly definite conditions. For instance, a turbine having a few wheels only, and using only a small difference of pressure, could be supplied with vanes having different angles, so as to determine the best angle for the vanes. Again, the spacing of the vanes could be altered, other things remaining the same, and, again, vanes of different widths could be tried. From tests on the completed turbine we ought to obtain, not merely consumption but the variation of pressure and temperature at different parts of the turbine, so as to compare with the corresponding quantities assumed in the design. In these and similar ways our knowledge of turbines would be extended, and better economy obtainable.

The Governing of High-Speed Engines.—I.

By J. H. DALES, A.M.Inst.C.E.

As high-speed engines are generally required to govern closely, and as the design of close governing entails a full knowledge of the principles on which governing apparatus is constructed, in dealing with this part of the subject it may be profitable to investigate those principles before proceeding to discuss the several types of governor which are in general use.

"Close governing" is generally considered to be such as is within 2 per cent. total variation, while ordinary governing may include such as is between 5 and 10 per cent. total variation; and, further, to make a distinction between the term "variation" as ordinarily used (which is too often in the sense of "deviation" from the normal), and "total variation"; the former means only half the actual or total variation of speed. Governing of some degree may be done by means of any physical effect which is brought about by a change of speed in the machine to be governed; but as "fine" governing is an extreme mechanical refinement, it can only be accomplished satisfactorily by those effects which are most free from and independent of friction and outside influences, or such as are absolutely constant. All such effects must be free forces and not controllable by anything but the principles on which they are applied. Applications of the effects of inertia are more reliable than any other effects of variation of speed, and are therefore generally made use of for purposes of governing, while the inertia of constant deviation is that which it is generally most convenient to use, and which is therefore commonly adopted. The inertia of constant deviation is, of course, synonymous with centrifugal force. The centrifugal force of a body depends upon its rate of deviation, namely—

$$\frac{v^2}{r} = \frac{\text{velocity}^2}{\text{radius}}$$

The deviating force = rate of deviation × weight $\frac{}{g} = \frac{v^2}{r} \cdot \frac{w}{g} = \frac{v^2 w}{r g}$ = deviating force

$= \frac{\text{velocity}^2 \text{ in feet per second} \times \text{weight (lbs.)}}{\text{radius in feet} \times 32{\cdot}2 \text{ ft.}}$

= deviating or centrifugal force in lbs.

The centrifugal force of 1 lb. at 1 ft. radius at one revolution per minute would therefore be—

A. $\frac{\text{circumference}}{60 \text{ seconds}}$ = velocity per second

$= \frac{1 \text{ ft. radius} \times 6{\cdot}2832}{60 \text{ seconds}} = \frac{6{\cdot}2832}{60}$

= ·10472 ft. per second = velocity (v).

B. $\frac{v^2}{r} = \frac{{\cdot}10472^2}{1}$ = rate of deviation in feet per second.

C. $\frac{v^2 w}{r g} = \frac{{\cdot}10472^2 \times 1 \text{ lb.}}{1 \text{ ft. radius} \times 32{\cdot}2 \text{ ft.}}$

= No. ·00034056 lbs.

The centrifugal force, therefore, of 1 lb. at 1 ft. radius at one revolution per minute is ·00034 (about) of 1 lb.; and, as it is seen that the centrifugal force of a body increases as the square of its angular velocity, directly as its weight and directly as its radius, the above ·00034 lb. may be taken as a constant for calculating the centrifugal force of a body, with *lbs.* weight, *feet* radius, and revolutions per *minute*, as units. So ·00034 × weight in lbs. × radius in feet × the square of the number of revolutions per minute = centrifugal force.

For purposes of rough preliminary estimation, it may be said that the variation of the centrifugal force of a governor to its total centrifugal force is, in proportion to

the latter, about double the percentage of the speed variation, so that 1 per cent. increase or decrease of speed gives about 2 per cent., more or less, difference of centrifugal force for governing purposes. This applies within ordinary ranges of variation of speed. Roughly, therefore, for every 100 lbs. centrifugal force of the governor, with a variation of 1 per cent., 2 lbs. of free force is generated; and it is evident that unless this very small amount is carefully handled in the arrangements it may be very easily lost in friction, or be hopelessly behind time from the effects of inertia. It is, therefore, easy to see that fine governing can only be done by governors having a very large centrifugal force and little or no friction of their own, or by such as are assisted by auxiliary power.

Fine or sudden governing can only be effected by an instant response to any difference of speed; and as inertia in such circumstances is by far the largest item of resistance to a change of position of the governing parts, it is absolutely essential to make all the moving parts which perform the act of governing with as little weight as possible. This, of course, refers to everything which has to be moved by the governor, both its own parts and any valves or expansion gears.

The logical conclusion from this is that, as the centrifugal force must be very great and the generating weights small, great speed of the governor is necessary to generate that force, and that loaded sliders are indispensable; but dead weight loading, having great inertia, being inadmissible, spring loads become a necessary feature in all fine governing.

One per cent. governing by dead-weight loaded governors is a chimera, so far as concerns sudden and large variations, such as occur in many phases of electric work, and which are liable to occur in many other services; but one per cent. governing can be done, by spring-load governors which are properly designed and applied.

Governors which will control an engine to one or one-and-a-half per cent., however, must be regarded as scientific instruments throughout their design and construction, in which there is absolutely no margin for guessing or rule of thumb.

The amount of force which a governor is required to exert varies from ounces to cwts., but is, in the majority of cases, a very difficult item to fix, and can seldom be directly ascertained until after a set of apparatus has been set to work, and then only by deductions from the actual results of working and the known centrifugal force of the governor. The principal items may, however, be ascertained by careful computation for the work to be done; while the centrifugal force provided should be as much as space and considerations of first cost of apparatus will admit of; always erring, if at all, on the liberal side, as nothing pays better than, or indeed so well as, good governing.

In considering high-speed governing it is scarcely worth while going into governors of the pendulous type, as all highly-loaded governors are unaffected except to a small extent by action of gravity on the generating weights; and the actions to be considered are generally only the mechanical relations of the different parts. The smaller matters can usually be provided for by adjustments.

The principal item for consideration in loaded governors is the spring which has to provide the load. The cylindrical spiral spring is the type which is generally used for loading of governors, but flat springs have also been applied, especially in American practice. It may be useful at this stage to investigate the nature of the stress in a cylindrical spiral spring. A cylindrical spiral is formed by winding an inclined plane round a cylinder, and a cylindrical spiral coil of true form and homogeneous quality, whether extended or compressed, maintains the true spiral position of its coils, extension or compression affecting the pitch and diameter of the coils only.

Assuming indefinite elasticity, if a cylindrical spiral be pulled out straight, without allowing of any axial rotation of the coils, the wire of which the spiral is formed would be twisted with one complete turn in each length of wire of which one coil was originally formed. Thus, if the coil was 1 in. diameter, centre to centre of

the wire, the fully-extended wire in the instance quoted would have one complete twist in 3·1416 ins., and so on; and the same resistance to torsion (theoretically) could be produced by twisting a straight wire of the length of the wire in the coil to the same extent.

The stresses in a cylindrical coil are produced in extension by strains which vary from a pure shearing and torsional action, in a coil of infinitely close pitch, to a stretching strain only, when the wire of the coil is drawn out perfectly straight; and the inference is that the exact conditions of stress or strain can be ascertained only for one point between the two extreme conditions. In a general way it may be said that the more a spring is stretched the stiffer (per inch) it becomes, while the more it is compressed the weaker (per inch) it is. These differences are sufficient to require consideration in load springs for close governing.

The writer never applies springs in tension for the purposes under consideration, on account of the risk of failure; the danger attached to the use of a spring in tension being generally greater than when used in compression. For the reasons given above, unless the differences of strength referred to are considered, a spring in compression, with a given initial compression, will allow a governor to act at a lower speed than that which has been calculated, to an extent which may sometimes be outside the range of adjustment, while the case of a spring in tension may also be inconveniently out of the calculation, as it will run its engine too fast. For about one per cent. governing, adjustments cannot be made for ranges of speed due to more than two to two-and-a-half per cent. of the loading; on account of their causing "racing" or "hunting" when made "up," or for an increase of speed; while letting out the adjustments to any extent spoils the fineness of the governing. It is therefore necessary to be very exact in reckonings for work of this kind. It must not, however, be supposed that after a good job has once been made there is anything delicate or difficult to maintain about it; on the contrary, nothing more than cleanliness and a reasonable attention to lubrication is necessary, and the results in governing rather improve than deteriorate as the fits wear easy. Good quality of spring is, of course, indispensable; nothing second-rate is of any value whatever, and the absolute best obtainable at any price is incomparably the cheaper. Also, as the exact strength of a spring is a matter of quality of material and temper, and its elasticity a matter of proportion of diameter of wire to diameter of coil, both of which vary with each other, no formula can be made to correspond with exact requirements, which latter can only be provided for and met by the manufacturer, and it is therefore necessary for the designer to ascertain what can be obtained in the way of a spring before providing accommodation for it. A spring of any given diameter of wire or of any given diameter of coil can, within practical limits, be made of any strength per inch (tension or compression) by increasing or decreasing the number of coils, but the ultimate stiffness of complete compression is the same whatever the length or number of coils may be, and that is the weight which is required for closing one coil. As standard rollings have generally to be used in the making of springs and also standard mandrels for the coiling, and the temper having to be as hard as will stand closing of the coils, the only means of adjustment to exact stiffness is in the way of altering the length; but this is a very simple matter. Of course a spring once made cannot have its stiffness reduced, as that would entail adding to the number of coils, at the same time maintaining the diameter; but by making the spring longer than the calculation, and consequently weaker than required, its exact strength can be ascertained experimentally, and it can then be proportionately reduced to give the exact requirements. High-class spring making, however, is no amateur job, and a knowledge of the nature of the process of production is of value to the designer of a governor only in so far as it gives him a better idea of what is possible, and therefore what he has to work upon and provide for. Briefly, he must have his springs first, and make his centrifugal

weights to give him the speed of rotation which he requires. Also, as the centrifugal effect of the weight of the spring upon itself, or as the weight of the spring generally forms part of the centrifugal weight, especially with extreme speeds—which also vary in performing the functions of the governor — the process of calculation is a much complicated one. A graphic method will, however, be shown for solving this part of the difficulty, at the same time providing an illustration of the action of a spring laid in a radius to the centre of the motion.

ROCK DRILLS.

By EWART C. AMOS, M.I.Mech.E.

WHILST it is not proposed in this article to make an historical survey of Rock Drilling Plant, it may prove interesting to briefly refer to some features of the earlier machines in order to compare them with the latest types as used at the present time. It is generally claimed by our American friends that the Rock Drill is essentially an American invention; and, although it might be difficult to sustain this claim in view of certain drills known to have been in use both in Cornwall, and also in Germany at a date previous to any kind of American mining, yet it is undoubtedly the case that at the present time our enterprising competitors from the West are largely controlling the markets in Rock Drills, and the demand for machines of their make is steadily on the increase. There are various reasons for this: in the first place they have in their own country a much larger demand for Rock Drills than we have here, which has led them to carry out very extensive experiments in order to determine the best form of drill to adopt. In the second place, having embodied their experience in the form of an excellent machine, they have put it on the market in a very thorough manner, and to a very large extent ousted our own makers, even where they are selling a machine which compares favourably with the imported article. From 1867 (and possibly earlier) and onwards there have been from time to time public trials of various types of Rock Drills, but it is a very open question whether they have produced any satisfactory results. When it is remembered that the conditions obtaining at such trials must of necessity be very different from those experienced in actual practice, it is easy to understand that the drill best suited to everyday work frequently does not show up well on these trials, especially when so much depends on the skill of the operator. Again, certain drills are suited to certain kinds of rock and under certain conditions; and to suppose that any reliable data can be obtained from setting to work a number of drills under conditions rarely met with in practical quarrying or mining operations, is likely to lead to considerable disappointment. Reference is made here to this point in order that intending purchasers of rock-drilling plant shall not be guided in their selection by such results. It is now generally accepted, by both makers and users alike, that a good drill should possess the following features, and the absence of such qualifications should condemn it except under very special conditions:—

1. It should be simple in construction.
2. The parts should be as few as possible, consistent with the economical use of the compressed air or steam which works it.
3. Every part should be strong and

capable of easy withdrawal and replacement in case of accident.

4. The striking piston should, as far as possible, take the entire force of the blow, and the sudden removal of the resistance should not cause injury to the machine.
5. The striking piston should have a variable stroke, and its rotation should be automatic.
6. The machine should be as compact and light as possible consistent with the above conditions.

In addition to these fundamental points, it should be econonical in its air or steam consumption, whilst the mountings on which it is set for different kinds of work should be capable of rapid and extensive adjustment. As in nearly all classes of machinery, each size and type of drill has its useful limit, both in respect to size of hole to be bored and kind of rock to which it is best suited; and any attempt to exceed this limit is not only unfair to the maker, but also entails disappointment and loss to the user. It may be said that the rock drill is the prototype of the portable pneumatic-hammer, which is so rapidly finding favour in most engineering works to-day; and it is largely due to the experience gained from rock-drill construction that has enabled the makers of the hand-tool to put upon the market a machine at once so perfect and simple, especially in its valve arrangements.

In the early rock-drills the valves were controlled by tappets projecting into the cylinders, and struck by the piston at either end of its stroke. This arrangement exposed both the valve and tappet to very violent shocks, and has been superseded entirely by valves regulated entirely or partly by the compressed air or steam, which is employed to work the drill.

In order to treat the subject in a comprehensive manner, it will be convenient to consider it under two headings—viz.: (1) Percussion Drills; and (2) Diamond Prospecting Drills.

Each of these classifications may again be subdivided into hand and power-driven drills. In the case of the percussion drills the motive agent is in the form of compressed air or steam acting direct on the striking piston, whilst the power-driven diamond-boring drills are fitted with engines or electric motors.

Percussion Drills.

By far the largest number of drills in use are of the percussion type. As is well known, the principle of machines for rock-drilling is based upon the results obtained in hand-drilling, viz. — a succession of blows given to a chisel, which is partially rotated between each blow. In hand-drilling the chisel is held in position and rotated by one man, whilst another strikes the blows; or, in place of a chisel, a jumper (a long heavy bar with a chisel end) is used by one man, who jumps it up and down a hole, and also rotates it. In hand-drilling, it will be seen, the blow is essentially a dead blow, and the best modern-practice drills now obtain the same result. In some of the earlier machines, however, the air for effecting the return stroke was admitted in front of the piston before the blow was actually struck. Most modern drills are of simple construction, and consist mainly of a shell or cylinder, which contains the striking piston, the extension of which forms the drill bit. The cylinder is carried on a slide, and this in turn is mounted on a tripod, column, shaft, bar, or other special mounting, to suit the various purposes for which the drill is to be used. There is also a valve for regulating the proper distribution of the working fluid at either side of the piston, and to exhaust, and a further mechanism for giving a rotation or partial rotation to the bit after each blow struck. This, together with a feed mechanism and stop-valve, practically comprises the whole of the working parts, and is common to all drills of this type, each type differing only in respect to the kind of valve and rotation motion employed. As to the sizes and capacities, the general sizes are drills having cylinders from 2-in. up to 5-in. bore, and with strokes from 4 ins. to 8 ins. Taking a 3-in. machine giving 350 blows per minute, this would have, say, 6-in. stroke, 24-in. feed, and by changing bits will

FIG. 1.—INGERSOLL-SERGEANT ROCK DRILL.

drill easily up to 15 ft. It will bore a 1¼-in. hole, and a fair average of work done in 10 hours in down holes in granite, including time taken for changing bits and setting drill, would be about 65 to 70 ft. It would require steam power equal to an 8-h.p. boiler (effective) to drive, would weigh, complete with tripod, weights, etc., about 6½ cwts., and costs about £70. These figures are approximate, but represent a fair average of the capacity, size and cost of a modern drill. With regard to the power required it must be noted that the h.p. given is actual and not nominal, and it cannot too strongly be urged upon users to have a surplus rather than a shortage of steam or air supply.

Coming now to a description of a few types of modern drills, reference must be had to the accompanying illustrations and sections. One of the best known of the imported drills is the Ingersoll-Sergeant, which, besides being one of the earliest, is also to-day one of the most efficient. Figs. 1 and 2 show respectively a general view and section of an Ingersoll-Sergeant auxiliary valve-drill. This type is specially designed for hard service in shaft sinking, heading work, etc, when rapid progress has to be made through hard ground. As an example of what it has done may be mentioned the remarkable progress of 345 ft. of 10 by 20 ft. heading in one month by two of these

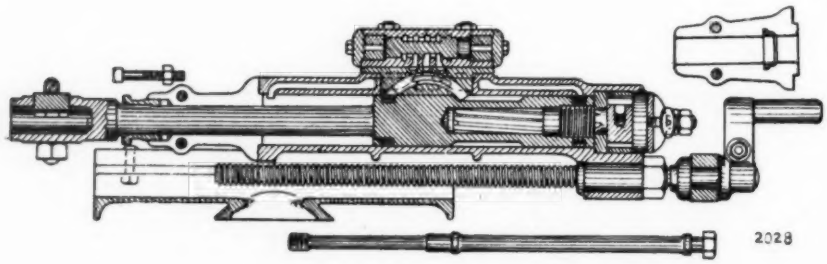


FIG. 2.—INGERSOLL-SERGEANT ROCK DRILL: SECTION.

machines. This machine strikes an uncushioned blow, as the pressure only enters the front end of piston after the blow has been struck; and it is made with two valves, which is its special feature. Reference to the sectional view will show that there is a round piston-valve working in a bushing on the side of the main cylinder, in a position common to all drills of this type. Suitable port-holes in the valve and bushing provide for the automatic admission of the steam or air to the cylinders, and these are so arranged as to secure a large opening with but a small movement of the valve. The piston also being balanced or cushioned at either end of its stroke, reduces breakage of valves to a minimum. After passing the piston or main valve the air or steam is further controlled by the auxiliary valve seen immediately under the piston valve (Fig. 2). This valve moves in the arc of a circle, and acts as a trigger to the main valve, and opens or closes the steam or air passages, releasing the pressure from one end or other of the main valve. It is very light, is made of steel, and being under constant pressure is kept well up to its seat. This arrangement ensures that the main valve is retained in such a position that while the piston carrying the cutting tool is moved towards the rock the exhaust remains open on one end, while the full pressure acts on the other until the blow is struck, at which time the valve is immediately released and reverses. No attempt is made in this drill to use the air expansively, as the makers consider that the advantage of getting the hardest possible blow from the smallest possible cylinder and least weight machine more than outbalances any saving effected by expanding the steam or air used; since when such is done a larger machine has to be used to give the same effective result, and this means time lost in moving the heavier machine. The auxiliary valve is worked by an excrescence formed on the side of the moving piston, this excrescence operating upon it tangentially at each stroke. In starting or blocking holes it is desirable that the drill should be able to make a short stroke. This variation in the length of stroke is obtained by turning the crank attached to the feed screw, and feeding the cylinder towards the rock. Another feature in this machine is the rotating device. The rotation of the piston and drill bit is probably one of the most crucial points in all rock drills, and one which in the past has been a cause of a great deal of trouble. Where several pawls are working on a ratchet-wheel in the cylinder-cover as in the old drills, it frequently occurs that, when the drill was not making its full stroke, the point of a pawl and the point of a ratchet-tooth were brought together in such a way as to let the ratchet-wheel step back a little, and the pawl was then apt to get a bit chipped off, and small pieces forced their way into the cylinder. Even with modern drills, when a hard blow is struck on an uneven surface, producing a tendency to twist the steel in the opposite direction to which it rotates, there is frequently a breakage of the rotating device. In the drill under description, the twisting of the spiral bar or breaking of parts or ratchet is avoided by the use of an internal ratchet-wheel with pawls forced out into the ratchet teeth by small spiral springs which have a straight-line movement, instead of working on a curve. Should the drill receive such a blow as described above, the effect is to simply turn the rotating ratchet around, overcoming the friction of the back-head springs, instead of twisting the rifle-bar, or breaking the pawls and ratchets. Fig. 3 shows this rotating device. Any blow upon the front or back end of the cylinder is cushioned by springs placed on the back head and connected to the front head by side bolts. The Ingersoll-Sergeant drills are made in a variety of sizes, and are fitted with other forms of valve gears, varying somewhat from the one already described.

Automatic Feeds.

Experience has shown that with the exception of the largest drills, an automatic feed has very little value, and in fact a hand-fed machine will do the more work. For large drills, however, engaged on surface work and in drilling "down" holes, it is of considerable importance, and

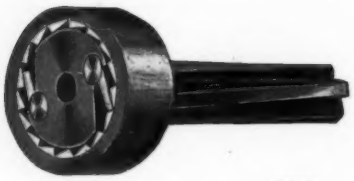


FIG. 3.—INGERSOLL-SERGEANT ROCK DRILL: ROTATING DEVICE.

effects an appreciable saving. The 4-in. and 5-inch "Ingersoll-Sergeant" drills are fitted with an automatic feed by which the cylinder is fed forward at each stroke of the piston. This is effected by means of a knuckle joint which enters the lower end of the cylinder, and is caused by the piston at the end of its forward stroke to turn the nut which feeds the cylinder forward. A friction strap provides for variation in feed according to the nature of the rock.

This is one of the best and simplest forms of automatic feed, and is in considerable use.

The next machine to be considered is the "Daw" drill. This is of British design and construction, and embodies some interesting features. This drill has been in use for a great many years, and has a well-established reputation as a good reliable machine. Figs. 4 and 5 show the drill in perspective, outside view, and also in section. A claim made for this machine is that it is one of the lightest and most compact drills made, and that its valve action is not impaired by leakage or wear. It may be said to largely fulfil the conditions already laid down as necessary in a good drill, and in possessing fewness of parts it certainly excels. It will be seen from the section that the valve consists of one part only, and is of the piston air-controlled type, and so controlled by the main piston that it is automatically locked by the fluid pressure after each stroke, besides being perfectly balanced. There is no dead centre, and the drill can be started in any position. This machine also gives a direct blow on the rock, and a rapid recovery is secured by the admission of live air at full pressure immediately after the blow has been struck. This is a useful feature when the ground is light or pockety. The speed as well as the blow can be regulated to the requirements with a maximum speed of 650 strokes per minute. This speed has been attained in Aberdeen granite. The weight of a 3-in. "Daw" drill complete

FIG. 4.—"DAW" ROCK DRILL.

with cradle, but without mountings or weights, is 230 lbs.

The makers of the "Daw" drills, like others, have their own special tripods, columns, clamps, etc., but these mountings are referred to under a separate heading.

Fig. 6 illustrates a No. 3 B "Rand" drill mounted on a stopping bar, as used for stoping and drilling in the head of a drift, where a number of holes are put in a row. The bar is fitted with a jack-screw and special lock-nut, the drill being mounted on the bar by a special clamp. The bars vary in length, but an ordinary length would be about 8 ft. In this drill the valve is a plain slide-valve, always thrown in the same direction in which the piston is moving, the opening and closing being effected in a positive manner. Motion is transmitted from the main piston to the slide valve by means of a three-arm locker or lever, the said lever being held in position by a pin. The rocker is placed in a recess in the cylinder, and the piston is caused to engage with it at each in-and-out stroke, and thus moves the valve to regulate the distribution of the actuating fluid.

Fig. 7 shows in section the valve arrangement of the "Slugger" type of Rand drill. In its general construction and apart from the valve, the "Slugger" drill is similar to the drill shown by Fig. 6. The valve is of the piston type, and besides being perfectly balanced, has a stroke of only $\frac{3}{4}$ in. In action it allows the striking piston to give an uncushioned blow, and at the same time uses the air expansively. The machine may be built to cut off early or late, according to requirement; so that, in districts where labour is cheap and fuel expensive, a considerable degree of expansion may be practised. Another feature possessed by this drill is that exhaust instead of live steam is used to secure a cushion to stop the piston on its return stroke. When air is used to drive this drill it is fitted with a special leather-packed lower head. This drill is designed for all-round work; it will strike a hard blow with a good pull back, whilst any wear on the piston increases rather than diminishes the stroke.

Fig. 8 shows a general view of a "Schram" drill mounted on a tripod. Its valve is of the piston-air-controlled type, and the machine embodies the qualifications of a good drill.

Fig 9 shows the section of the "Optimus" drill, which is made by the same maker as the "Schram" drill. This machine has met with considerable success, and possesses some novel features, distinct from most of the modern drills. It works on the compound principle, which effects a considerable saving in the consumption of the actuating fluid. It will be seen by reference to the diagram that the cylinder and piston have two diameters, and the automatic valve provides, on the forward stroke of the piston, for admission of air under full pressure to the upper end of the cylinder, whilst the air is simultaneously being exhausted from the lower end. Further, the air which has been used to make the forward stroke, instead of being exhausted to the atmosphere (as in most drills), is taken through the valve to the under side of the lower or larger piston, and is utilised again for the backward stroke. Great economy is thus effected in the use of air and consequent first cost of plant and subsequent

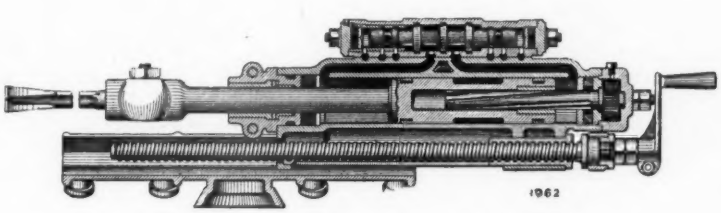


FIG. 5.—SECTION THROUGH "DAW" ROCK DRILL.

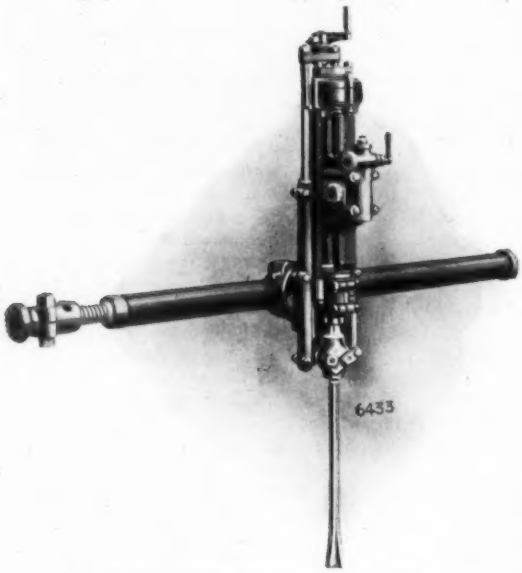


FIG. 6.—"RAND" DRILL MOUNTED ON STOPING BAR.

maintenance. The operation of the machine is as follows:—

Assuming the piston *e* and valve *f* to be in the position shown in the illustration, the cylinder *a* will be in communication with the air under pressure through the port *b* (the cylinder a^1 being at the same time in communication with the atmosphere through the ports *m* and *h*); the result of this will be that the piston is forced forward, and immediately the piston *c* uncovers the small port *d*, a portion of the compressed air passes into the small cylinder *r* behind the valve-piston *e*, and acting on a larger area than that which is subject to the constant pressure at *l*, and the fact that there is no resistance at the other end of the valve-piston *e*, owing to the passage *n* communicating with the atmosphere, forces the valve *e* over to the other end of the valve chest, thereby cutting off communication with the air under pressure, and placing the cylinders *a* and a^1 in communication through the ports *b* and *m*. Then the air that has acted on the piston *c* now passes into the cylinder a^1, where it acts on the piston *g*, of larger area than the piston *c*, thereby moving the piston backwards to its original position. When the piston *c*, during its backward stroke, passes the port *d*, the cylinder *r* is placed in communication with the atmosphere through the ports *d* and *h*, and the constant pressure acting on the piston-valve *e* at *l*, moves the valve *f* over to the position as shown in the illustration; the air again enters the cylinder *a*, and the action is repeated.

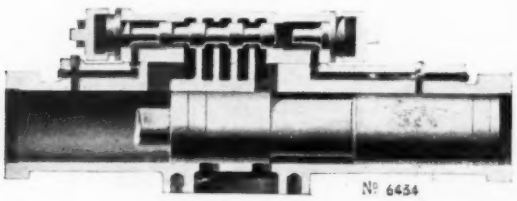


FIG. 7.—SECTION OF VALVE ARRANGEMENT "SLUGGER" TYPE OF RAND DRILL.

Drill Mountings.

These form a very important feature in a drill outfit, and are the subject of various patents. Different makers recommend their own specialities. The tripod, in some shape or form, is common to them all, and is used for general out-of-door work, being capable of very great adjustment, and allowing the drill to be fixed in all kinds of positions, including lying on the ground or close against a wall. A "Lewis" hole tripod permits of several holes being drilled without moving the tripod, by providing a short slide to which the three legs are attached, and on which the cradle of the drill can be adjusted.

Quarry bars are developments of the Lewis tripod, and will carry several drills on the same bar. They are fitted with racking and other motions. Columns and shaft-bars are fixed vertically or horizontally, and any angles, by being forced up by a jack-screw between the sides or top and bottom of a shaft. The drill is mounted upon them by an adjustable clamp, and swivel arms afford further adjustability. There are, in addition to these, a large number of special mountings or frames fitted to trolleys or carriages with self-propelling motions for channelling, etc.

Thus far, only power machines, driven either by steam or compressed air, have been dealt with. There is, however, another type of percussion drill to which brief reference must be made, viz. the hand-power drill. There exists considerable difference of opinion as to the value of hand-power drills, and makers of rock drills always recommend the power machines wherever they can possibly be used. At the same time, however, there are occasions where it is not possible or convenient to put down steam or air supply, and even in large mines where such plant exists they are very handy for special purposes. It is true that some of the earliest machines did not compare very favourably with hand-drilling, but some of the modern drills certainly save considerably over this method.

Fig. 10 shows an "Ingersoll-Duplex" hand-power drill being worked by two lads. It is made in two sizes, the smallest size

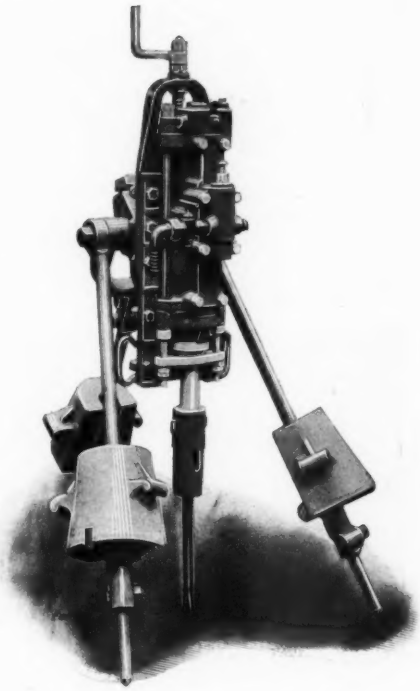


FIG. 8.—"SCHRAM" ROCK DRILL.

requiring but one lad to work it. It is really a mechanical hand-jumper, and the principle upon which it works will readily be seen by reference to the illustration. The drill-bit and holder are attached by

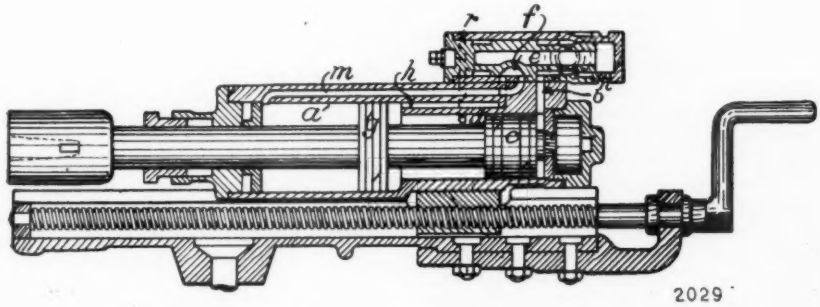


FIG. 9.—SECTION OF "OPTIMUS" DRILL.

connecting-rods to a double-crank shaft, on which are mounted the flywheels and handles. By revolving the latter, powerful springs (attached to the framing and the drill-holder) are compressed and released, and the blow is thereby struck. The rotation of the bit is automatic and positive, so that it never strikes the rock twice in the same place. One of the most important features in this drill is that by one revolution of the flywheel two or more strokes of the bit are made without any material tax upon the power applied to

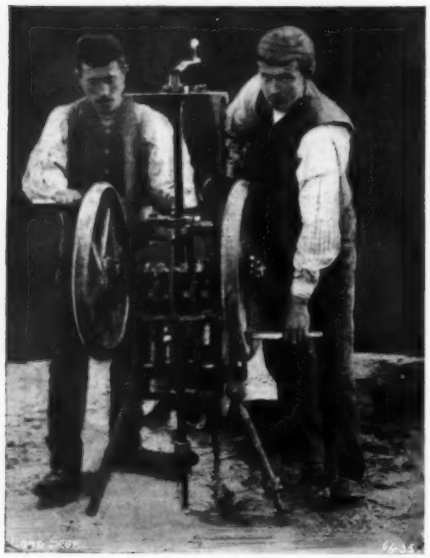


FIG. 10.—INGERSOLL-DUPLEX HAND-POWER DRILL.

turn it. It can be worked at any angle on a horizontal or vertical surface, and, when necessary to bore at a higher point than the legs will allow, it can be used on a frame in any position desired. The drill can be rolled easily on its wheels by one man on surfaces sufficiently level to admit the use of a wheelbarrow; and, the wheels being easily and quickly removed, one or two men can carry the body of the drill. A drill, as illustrated, weighs 3 cwts., and will strike blows of 350 lbs. each. It can be run at about 140 blows per min., and will bore 1-in. holes in granite at the rate of about $1\frac{1}{2}$ in. per min.

Before coming to a description of diamond drills, which constitute the second heading, as set forth in the early part of this article, reference must be had to the "Humboldt" (Trantz patent) rotary drill, shown by Figs. 11 and 12. This machine is specially suited for soft minerals, such as slate, coal, rock-salt, etc., although it will compare very favourably with hand-drilling on still harder materials. The machine consists principally of a wrought-iron frame, to which are attached two oscillating cylinders, each $3\frac{3}{8}$-in. diameter by 2-in. stroke. The piston actuates a two-throw crank and two sets of double-geared spur-wheels. These wheels are used to turn the drill-spindle, and at the same time to give the feed, and the machine is so arranged that the speed of rotation of the bit and feed may be suited to the hardness of the material to be drilled. The methods of support and attachment to the column are shown by the illustrations. This machine, which is of German design and construction, can be worked at a maximum speed by either compressed air or hydraulic pressure at 35 lbs. to 45 lbs. per sq. in.

Diamond Prospecting Drills.

A full consideration of this type of drill would require further space than this article permits, and it is only possible, therefore, to deal with them in a brief manner; on the other hand, to omit reference to them entirely would make the article incomplete. Although the diamond drill does not compete with the percussion drill for excavating purposes, owing to its being more expensive in working, yet it has functions outside the percussion drill which make it invaluable in certain classes of mining. Its chief use is to secure a sample of the material bored through, and this it does by using a hollow bit, which leaves a core of the rock or other substance bored through; hence in ascertaining the nature of the material it is of great assistance to

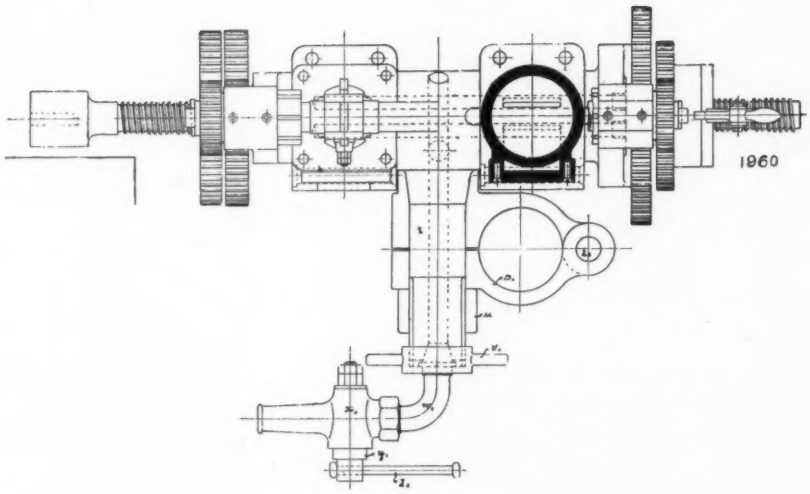


FIG. 11.

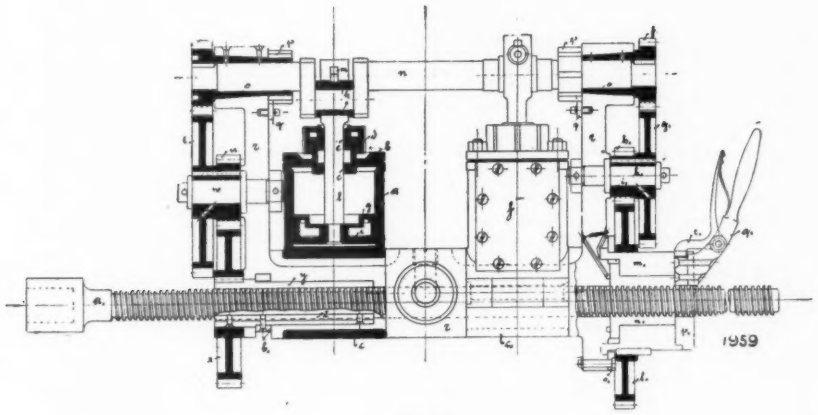


FIG. 12.

FIGS. 11 AND 12.—SECTIONS OF HUMBOLDT (TRANTZ PATENT) ROTARY DRILL.

the prospector or miner, and enables him to determine whether he shall excavate in any particular spot. Diamond drills are made to be driven either by hand or power, some of the latter type being of very elaborate and powerful construction. The principle upon which they work is the rotation of a hollow spindle carrying a special hollow bit fitted with diamonds or carbons. One of these bits is shown by Fig. 13. The rotation of these, cuts or grinds away a ring of the material, leaving in the hollow spindle a core of same, which is brought to the surface for examination from time to time during the progress of the boring.

Dealing with the hand-drills first, Fig. 14 shows the "Schram" type. This

FIG. 13.—DIAMOND DRILL BIT.

machine is of very simple construction and can be easily handled. It is also capable of being driven by power. The bevel gearing, by means of which the spindle is driven, is enclosed in a case, whilst the feed is maintained by the rack and pinion with weight seen in the top of the machine. Water is fed down the centre of spindle by means of the hand-pump shown.

This machine will bore a hole 400 to 500 ft. deep at the rate of about one foot per hour in hard material.

Fig. 15 shows a "Sullivan" hand-power drill of American construction. The frame is tubular, and the machine is fitted with a drum and wire rope for pulling out the drill rods. The method of driving and feeding the drill rods is clearly shown in the illustration. It has a capacity up to 300 ft. deep, and will drill a hole $1\frac{5}{16}$ ins. diameter, giving a core of $\frac{15}{16}$ in. diameter.

Fig. 16 shows a "Sullivan" power-drill driven by an electric motor, and fitted complete with a pump and hoisting drum. This machine has the same drilling capacity as the hand-machine just described. Some of the largest sizes of diamond-drills are constructed to bore holes one mile deep and $3\frac{3}{4}$ ins. diameter, with a core of $2\frac{3}{4}$ ins. The rods in such a machine would weigh 20 tons, and it is easy to calculate that were they made of uniform section they would pull apart by their own weight. This difficulty is, however, overcome by successively reducing their size as the depth of the hole increases, making, in effect, a taper rod.

Air Consumption of Rock Drills.

Some reference to this may prove interesting taken into consideration with the relative cost of steam and compressed air; the figures being based on pressures at sea level. Dealing with air-consumption first, and taking a 3-in. drill, a fair average consumption would be 100 cub. ft. of free air per minute at 60 lbs. pressure. This amount has, of course, to be increased

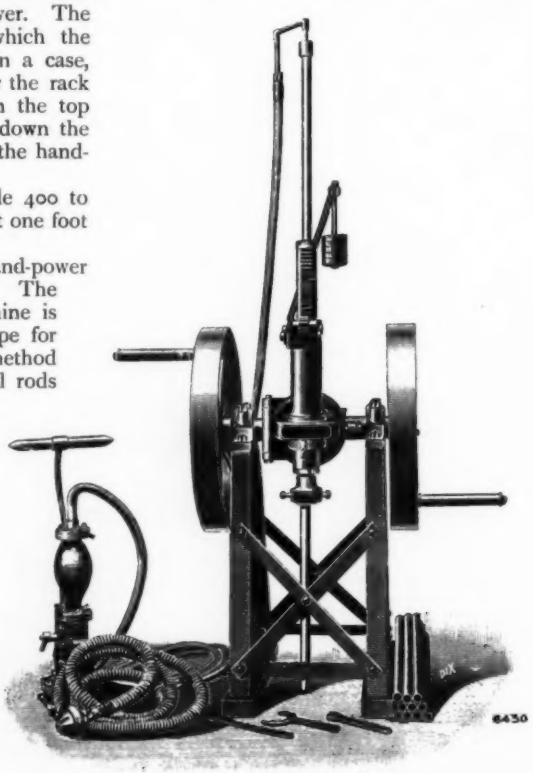


FIG. 14.—"SCHRAM" DIAMOND PROSPECTING HAND-DRILL.

according to the altitude at which the drill is working above sea level. Compressed air as the motive power has many advantages over steam, although the latter is much used, especially where steam plant exists for other purposes. In quarry work, where operations have to be carried on at considerable distances apart, the loss in condensation is very great when using steam. Again, an air-driven drill is far more convenient to operate than a hot steam drill, whilst the hosing lasts much longer when air is used. Besides these advantages, practice has now proved that more work can be done at less cost with the air-driven plant, and actual tests have shown the saving to be at least 25 to 30 per cent.

In conclusion, there is, in the author's opinion, very much larger scope for the use of rock drills in this country than has up to the present obtained; in many of our slate and stone quarries the old system of hand drilling is still in vogue, and even where drills have been adopted they are frequently found to be of old design and often discarded, no attempt being made to keep them in working order, or to replace them by more modern machines. Miners in this country are somewhat conservative in their business habits, and it becomes necessary for the manufacturers to press home the advantages of this matter.

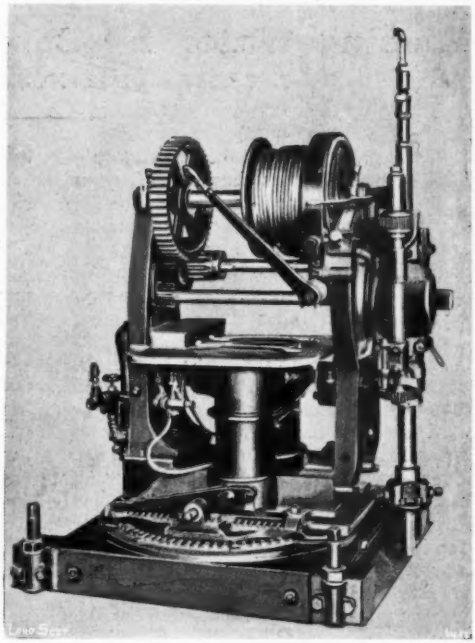


FIG. 16.—ELECTRICALLY-DRIVEN "SULLIVAN" POWER-DRILL.

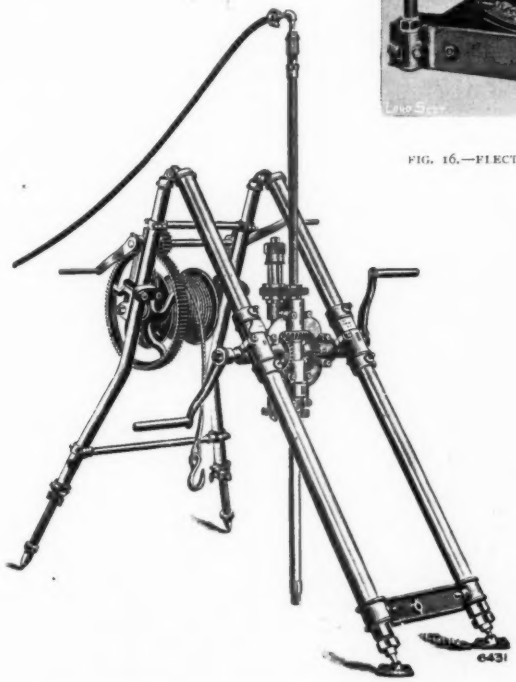


FIG. 15.—"SULLIVAN" DIAMOND PROSPECTING HAND-POWER DRILL.

Raising Water by Compressed Air.

By C. T. ALFRED HANSSEN, Assoc.M.Inst.C.E.

THE great interest taken in this subject, which has shown itself by an animated correspondence in the columns of the weekly technical press lasting nearly six months, is amply justified by the success obtained with this system in lifting large volumes of water, petroleum, and other fluids from great depths and to considerable heights.

Since the first introduction of the air-lift pump by Dr. Pohlé in the United States in 1893, it has been extensively used there both for raising water and for raising petroleum from bore-holes. They were introduced to this country by Mr. Geo. Y. Murray of Queen Victoria Street, who claims to have put down over 300 installations; and air-lift pumps are now made by a number of English firms. On the continent Mr. A. Borsig claims to have put down 600 installations of the special form of air-lift pump patented by him under the name of "Mammut" pump.

Among the chief advantages of the air-lift system may be mentioned that it obviates the necessity of putting down expensive installations of deep-well pumps, with their large and costly wells. Most engineers know the difficulty of keeping such deep-well pumps in repair, and also their low efficiency, which is chiefly due to the friction of the rods in their guides and the difficulties of getting all parts of the machinery correctly aligned. The air-lift pump can be put down in an ordinary bore-hole, and in many cases the bore-hole itself can be used as a rising-pipe by inserting a central air-pipe. Fig. 1 shows the simplest form of air-lift pump, where the whole extra cost consists of the central air-pipe, provided with a suitable air-nozzle *n*, for distributing the air in the water at the required level. There is, therefore, an entire absence of moving parts below the ground level, and a corresponding freedom from repairs and stoppages. As the compressed air in an air-lift pump works expansively, and is heated by the water during expansion so as to expand isothermally, a good efficiency can be obtained, and the cost of working compares favourably with that of a deep-well pump for the comparatively shallow lifts over which reliable experiments extend, and there can be no doubt that for deep immersions and high lifts the efficiency of an air-lift pump increases very much if proper precautions are taken.

In many cases water has been obtained from bore-holes by means of air-lifts where previous attempts at getting water by pumping have failed, owing to the presence of running sand in the water-bearing strata. Such sand quickly fills the well and plugs the bore-hole; but the air jet of an air lift pump is most efficient in clearing out sand and washing it away with the water.

The thorough aëration of the water raised by air-lift pumps is, as a rule, most beneficial; for although the presence of air in the water does not, as formerly supposed, destroy organic matter directly, it nevertheless does it indirectly by supplying oxygen to aërobic microbes. It may also act chemically on the water—for instance, by oxidising the soluble monoxide of iron so often found in deep-well water, and by transforming it into the insoluble dioxide of iron, which can be removed by filtration.

Fig. 2 shows the general arrangement of the "Mammut" pump used by Prof. E. Josse for his experiments published in *Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 3rd, 1898, to which reference has been made in columns 1 to 9 of Table I. Fig. 3 shows a detail of the same pump, the air-inlet connection from the air-pipe to the rising-pipe of the air-lift pump Figs. 4, 5, and 6 show section, end-view, and plan of air-lift pump for sewage effluent put down by Messrs. Hughes & Lancaster, according to the design of Messrs. Shone & Ault, C.E. and M.E.,

at the Sewerage Works at Hampton-on-Thames. In this case it was required that the air lift pump should adjust itself automatically to the varying flow of the effluent, and this is done by means of a bell-and-cup arrangement *b* and *c*, which actuate an air-valve *v*, so as to cut off the air supply when the water-level falls below *c*, and to open the valve when the water rises to *b*.

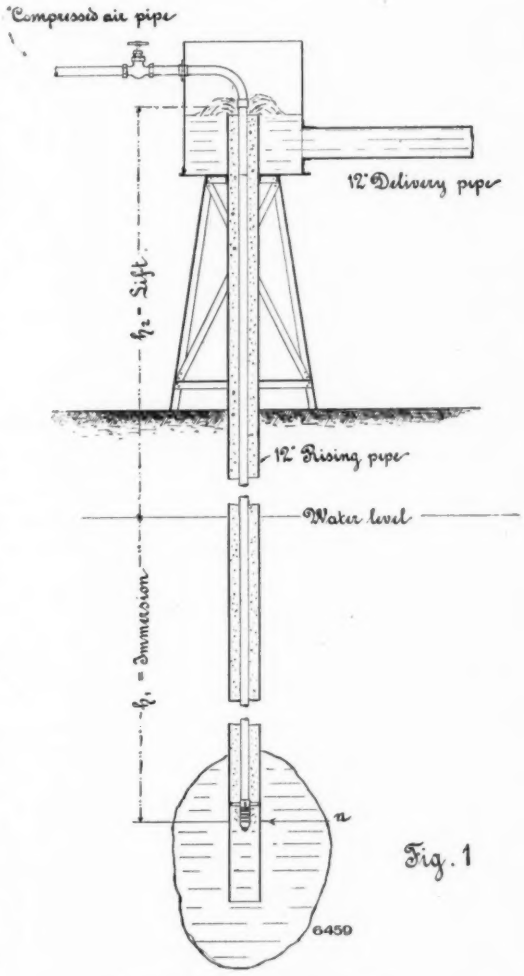


Fig. 1

FIG. 1.—SIMPLE FORM OF AIR-LIFT PUMP.

Theory of the Air-lift Pump.

The action of the compressed air in the rising pipe differs from that of the working-fluid in all other forms of lifting apparatus. In pumps, pulsometers, or ejectors the steam or compressed air acts either indirectly or directly by pressure; in injectors the steam acts by its velocity and momentum, but in the air-lift pump the compressed air acts entirely by its volume. The pressure of the air does not depend upon the height h_2 (Fig. 1) to which the water is lifted, but upon the depth of immersion h_1; that is to say, the air pressure must be sufficient to overcome the pressure of the liquid to be raised, as otherwise the air would not be able to flow out through the air-nozzle at the point *n* and mingle freely with the liquid in the rising-pipe. In doing this the air and water form a mixture with a specific gravity lower than that of water, which, so to say, floats on the outside column of solid water. It is, therefore, this latter column which causes an upward movement of the mixture of air and water in the rising-pipe.

During the upward movement of this mixture the volume of the compressed air gradually expands as it approaches the outlet, owing to the decreasing pressure of the liquid; and as this expansion of the air takes place in the presence of a relatively very much greater weight of water, from which it can absorb sufficient heat to maintain its temperature practically uniform, it is evident that the air must expand isothermally.

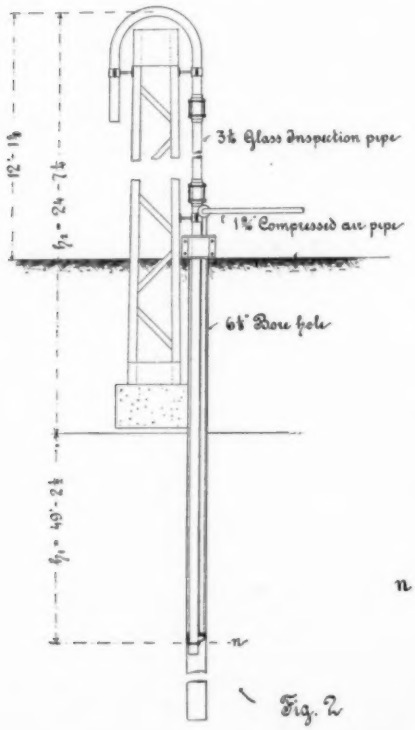
As air-bubbles rise through water with

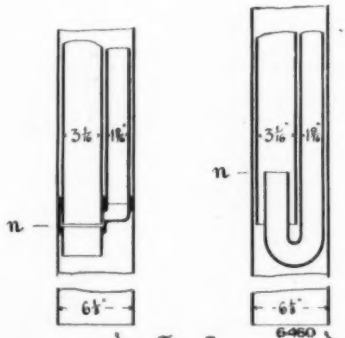
a considerable velocity it is necessary that the upward movement of the mixture of air and water should take place with the same velocity, as otherwise air will escape without doing work. This is shown in lines 13 and 19 of Table I.; the excess of air used above the theoretical quantity is very large for all the mean velocities below 8–9 ft. per second. It decreases as this velocity is reached, and again increases for very high velocities, as those in columns 9 and 12. The low efficiency of the Hampton air-lift is thus entirely due to the low velocity of the ascending column of air and water; in the experiment column 15 over 92·2 per cent. of air escapes from this cause. The series of experiments carried out by Prof. Josse are very instructive on this point. By comparing line 19 with line 8 it is seen that there is no increase in the quantity of water raised when the mean velocity exceeds 12·5 ft. per second; while at 22·8 ft. per second there is actually a decrease in the quantity of water, as the air is evidently blown through the rising-pipe in large bubbles without doing any useful work.

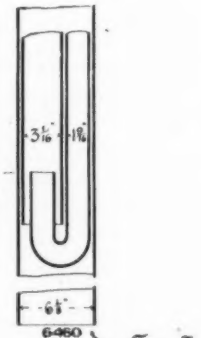
By comparing lines 9 and 13 of Table I. with line 24 it is seen that although the excess of air used with a mean upward velocity of 6·64 ft. per second (column 2) is much greater than with a velocity of 9·46 ft. per second (column 4), still, the efficiency in the latter case is much lower, owing to the greater frictional resistance at this velocity. Owing to the high velocity necessary for economical results the friction increases very rapidly, more especially for pipes of small diameters. In the Rotherham air-lift (column 10), where the frictional resistance is low by reason of the large diameter and short length of the rising-pipe, the conditions are very nearly theoretically perfect, the excess of air used being only $4\frac{1}{2}$ per cent., and an efficiency of nearly 50 per cent. is obtained.

Quantity of Free Air required for any given Air-lift Pump.

The theory of Dr. Pohlé, the inventor of the air-lift pump, as explained in his

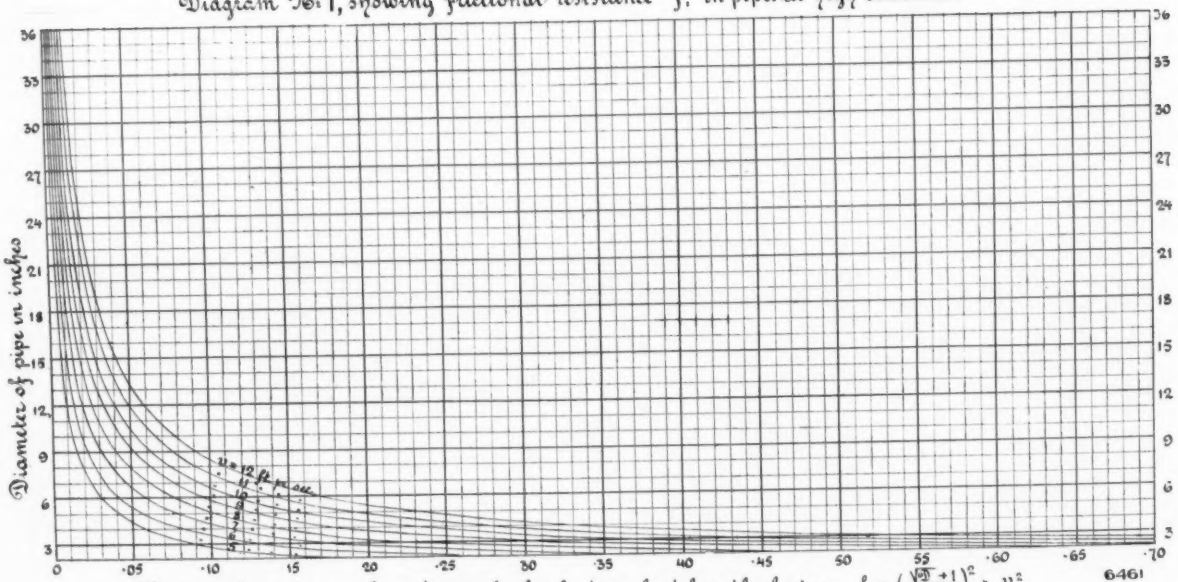






FIGS. 2 AND 3.—GENERAL ARRANGEMENT AND DETAIL OF "MAMMUT" PUMP.
FIG. 7.—POHLÉ'S PLAIN-AIR DISCHARGE-PIPE.

Diagram No 1, showing fractional resistance · f. in pipes at high velocities.



Fractional resistance f. in decimals of a foot per foot length of pipe $f. = \left(\frac{\sqrt{d}+1}{100\sqrt{d}}\right)^2 \cdot v^2$

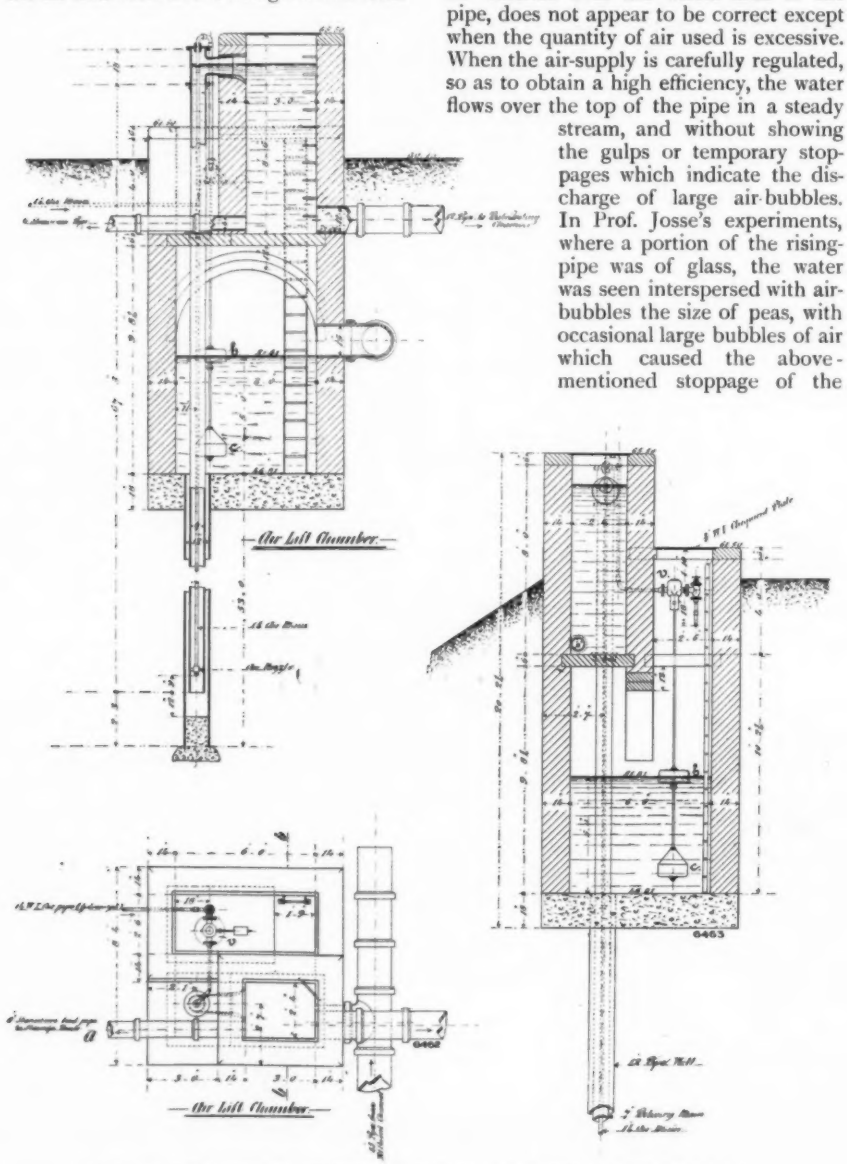
Table I.—Giving Results of

			1	2	3	4	5
		AUTHORITY AND PLACE OF TEST ...	Professor E. Josse,				
		DESIGNER OF PLANT & MANUFACTURER	A. Borsig, Berlin.				
		DESCRIPTION OF PLANT:					
1		Diameter of rising-pipe (ins.) ...	$3\frac{1}{16}$	$3\frac{1}{16}$	$3\frac{1}{16}$	$3\frac{1}{16}$	$3\frac{1}{16}$
2	a	Free area of ,, (sq. ft.)..	0·05115	0·05115	0·05115	0·05115	0·05115
3		Dia. of air-pipe, external (ins.)..	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$
		,, ,, internal ,, ...	...	...	...	...	...
4		,, outside tube ,, ..	$6\frac{1}{8}$	$6\frac{1}{8}$	$6\frac{1}{8}$	$6\frac{1}{8}$	$6\frac{1}{8}$
5	h_1	Depth of immersion (ft.) ...	49·21	49·21	49·21	49·21	49·21
6	h_2	Dead lift (ft.)	24·61	24·61	24·61	24·61	24·61
		QUANTITY OF WATER LIFTED:					
7	G	Gals. per min.	24·21	68·23	80·34	85·84	93·76
8	Q	Cub. ft. per sec.	0·0648	0·1766	0·2149	0·2296	0·2508
		VOLUME OF AIR USED:					
9	Vo	Cub. ft. per sec.	0·1271	0·3096	0·4685	0·4844	0·7387
10	$Va = \frac{Vo}{Q}$	Cub. ft. per 1 cub. ft. of water	1·96	1·75	2·18	2·11	2·94
11	$Vm = \frac{Va}{1 + \log. \frac{p_1}{p_0}}$	,, ,, at pm ...	1·030	0·922	1·146	1·110	1·547
12	Va_1	$= 1 \times \frac{h_2 + f}{h_1} \times \left(1 + \log. e \frac{p_1}{p_0}\right)$	1·21	1·39	1·76	1·81	2·48
13	$\frac{Va - Va_1}{Va_1} \times 100$	Per cent. excess of air used % ...	62·0	26·0	24·0	16·5	18·5
		PRESSURE OF AIR:					
14	$p_1 - p_0$	Gauge pressure (lbs. per sq. ft.)	21·5	21·5	21·5	21·5	21·5
15	p_0	Pressure of atmosphere absolute (lbs. per sq. ft.)	14·7	14·7	14·7	14·7	14·7
16	p_1	Receiver pressure (lbs. per sq. ft.)	36·2	36·2	36·2	36·2	36·2
17	pm	Mean pressure = $p_0\left(1 + \log. e \frac{p_1}{p_0}\right)$,, ,,	27·95	27·95	27·95	27·95	27·95
		VELOCITY OF FLOW:					
18	$\frac{Q}{a}$	Water at entry (ft. per sec.) ...	2·54	3·46	4·20	4·49	4·90
19	$\frac{Q}{a} \times (1 + Vm)$	,, and air at pm ,, ...	5·16	6·64	9·02	9·46	12·46
20	f_1	Friction per ft. of pipe (ft.) ...	0·090	0·153	0·283	0·301	0·540
21	f	$= f_1 (h_1 + h_2)$,, ...	6·7	11·3	20·9	22·2	39·8
		POWER DEVELOPED:					
22	$P.HP.$	In water lifted $= \frac{Ci \times h_2}{3{,}300}$ ...	0·181	0·495	0·601	0·643	0·701
23	$I.HP.$	$= V_0 \times 24{\cdot}1\left[\left(\frac{p_1}{p_0}\right)^{\frac{1}{3{\cdot}2}} - 1\right]$ steam cylinder	0·61	1·47	2·23	2·31	3·51
24	$\frac{100\ P.HP.}{I.HP.}$	Useful Effect %	29·9	33·7	26·9	27·8	20·0

Trials with Air-lift Pumps.

	6	7	8	9	10	11	12	13	14	15
	Charlottenburg, Berlin.				Hughes & Lancaster, Rotherham	W. H. Maxwell, C.E., Tunbridge Wells.			Shone & Ault, C.E., Hampton-on-Thames.	
					Hughes & Lancaster, Ruabon.					
	$31\frac{1}{16}$	$31\frac{1}{16}$	$31\frac{1}{16}$	$31\frac{1}{16}$	12	7	7	7	7	7
	0·05115	0·05115	0·05115	0·05115	0·7854	0·2182	0·2182	0·2182	0·2524	0·2524
	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$	3	...	$2\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$
	...	...	...	...	48	15 & $13\frac{1}{2}$	15 & $13\frac{1}{2}$	15 & $13\frac{1}{2}$	12	12
	$6\frac{1}{4}$	$6\frac{1}{4}$	$6\frac{1}{4}$	$6\frac{1}{4}$	25·00	252·0	208·0	237·4	58·0	58·0
	49·21	49·21	49·21	49·21	9·50	115·0	159 0	124·6	13·0	12·75
	24·61	24·61	24·61	24·61						
	96·84	96·84	96·84	88·04	1808·6	401·7	329·8	438·2	440·0	330·0
	0·2590	0·2590	0·2590	0·2533	4·823	1·072	0·880	1·168	1·173	0·881
	0·8423	0·9300	0·9536	1·7658	3·441	2·893	7·450	4·342	1·228	0·875
	3·25	3·59	3·68	7·50	0·713	2·70	8·47	3·72	1·045	0·994
	1·710	1·888	1·937	3·950	0·446	0·865	2·850	1·205	0·344	0·357
	2·81	3·06	3·15	5·65	0·682	2·29	5·09	3·11	0·613	0·522
	15·6	17·3	16·8	32·7	4·5	18·9	66·5	19·6	70·5	92·2
	21·5	21·5	21·5	21·5	12·0	109·0	91·0	103·0	30·0	26·2
	14·7	14·7	14·7	14·7	14·7	14·7	14·7	14·7	14·7	14·7
	36·2	36·2	36·2	36·2	26·7	123·7	105·7	117·7	44·7	40·9
	27·95	27 95	27·95	27·95	23·5	46·10	43·80	45·40	31·10	29·80
	5·07	5·07	5·07	4·61	6·15	4·92	4·02	5·36	4·65	3·49
	13·71	14·62	14·87	22·80	8·90	9·16	15·45	11·80	6·25	4·74
	0·652	0·745	0·770	1·650	0·0315	0·189	0·535	0·314	0·054	0·031
	48·2	55·1	56·8	121·6	1·09	69·3	196·5	115·2	3·8	2·2
	0·727	0·727	0·727	0·660	5·21	14·25	15·88	16·56	1·735	1·273
	4·01	4·42	4·54	8·40	10·51	33·0	78·3	48·2	7·37	4·75
	18·1	16·4	16·0	7·9	49·5	43·2	20·3	34·4	23·6	26·8

patent specification No. 22,373, 1893, that the air and water in the rising-column form alternate layers, and that each bubble of air extends over the whole area of the pipe, does not appear to be correct except when the quantity of air used is excessive. When the air-supply is carefully regulated, so as to obtain a high efficiency, the water flows over the top of the pipe in a steady stream, and without showing the gulps or temporary stoppages which indicate the discharge of large air-bubbles. In Prof. Josse's experiments, where a portion of the rising-pipe was of glass, the water was seen interspersed with air-bubbles the size of peas, with occasional large bubbles of air which caused the above-mentioned stoppage of the



FIGS. 4, 5, AND 6.—SECTION, END VIEW, AND PLAN OF AIR-LIFT PUMP AT SEWERAGE WORKS, HAMPTON-ON-THAMES.

flow of water. By testing the plain-air discharge-pipe used by Dr. Pohlé (Fig. 7) against the air-nozzle of the "Mammut" pump under otherwise identical conditions, Prof. Josse found that the water lifted with this latter nozzle was 25 per cent. greater than with the former, and this he correctly attributes to the more complete mixture of the air with the water effected by the "Mammut" air-nozzle.

The conditions of maximum efficiency are, therefore, that the air-bubbles should be small and intimately mixed with the water, that the air and water should ascend with approximately the same velocity, and that the air, after having done its work, should be able to escape without encountering any useless resistance which would increase the terminal pressure of the escaping air above that of the surrounding atmosphere. Any such increase in the terminal pressure would require a corresponding increase in the initial pressure, and hence also in the volume of free air required for the air-lift pump.

The above-mentioned correspondence in *Engineering* has turned chiefly on the dynamical theory of the air-lift pump, the purely mathematical aspect being ably set forth by Mr. C. C. Mason of Preston. It is, however, evident that the dynamical theory is somewhat complicated by the effect of the potential energy of the air and the interchange of heat from the water to the air, and that it can hardly be satisfactorily solved without experimental research. However interesting this question is for the scientific inquirer, it is not essential for the practical engineer who is anxious to know the volume of free air required under any set of conditions, so as to put down an air-compressor just large enough to supply this quantity of air.

The volumetric problem of the air-lift is fortunately comparatively simple, for it is evident that in order to get a volume of water Q to balance a volume of water mixed with air $Q + V_m$, these two volumes must be proportional to the length of the pipes containing them. In order to produce motion the volume of air at the mean pressure p_m must be increased until the specific gravity of the mixture $\left(\frac{Q}{Q + V_m}\right)$ is sufficiently low to overcome the frictional resistance in the pipe, which may be denoted f. This may be expressed in the following equation:

$$\frac{Q}{Q + V_m} = \frac{h_1}{h_1 + h_2 + f} \quad \ldots \quad (1)$$

where—

Q = the volume of water or other liquid to be lifted in cubic feet per second.

V_m = the volume of air at the mean absolute pressure p_m due to isothermal expansion from the initial absolute pressure p_1 to the final absolute atmospheric pressure p_0, where

$$p_m = p_0 \left(1 + \log_e \frac{p_1}{p_0}\right).$$

h_1 = depth of immersion of the pipe (Fig. 1) from the natural water-level to the level of the air-nozzle in feet.

h_2 = height to which the water is lifted from the natural water-level in the ground to the discharge level.

f = frictional resistance in the pipe, expressed in feet-head of the mixture of air and water.

In equation (1) h_1 expresses the column of solid water corresponding to Q; while $h_1 + h_2 + f$ expresses the height of the column of mixture of water with air which can be balanced by the column h_1, and which corresponds to the volume $Q + V_m$.

By solving equation (1) the value of V_m is found

$$V_m = \frac{Q\,(h_2 + f)}{h_1} \quad \ldots \quad (2)$$

It is, however, desirable to find the volume of air at atmospheric pressure and mean temperature, as the size of the air-compressing cylinder is proportional to this volume. According to the well-known laws for isothermal expansion the volume of air is inversely proportional to the pressure, and the volume V_0 at atmospheric pressure p_0 is

$$V_o = V_m \times \frac{p_m}{p_o} = V_m \frac{p_o\left(1+\log_e \frac{p_1}{p_o}\right)}{p_o}$$

$$= V_m \left(1 + \log_e \frac{p_1}{p_o}\right)$$

Formula (2) therefore becomes

$$V_o = \frac{Q(h_2+f)}{h_1} \times \left(1+\log_e \frac{p_1}{p_o}\right) \quad (3)$$

In Table I. line 12 has been calculated according to this formula when $Q = 1$; while line 10 gives the actually measured quantity of free air per cubic foot of water raised. It has already been pointed out that the velocity of the ascending column of air and water has a most important influence on the quantity of air used, and a close study of these two lines is therefore most instructive.

Frictional Resistance in the Rising-pipe.

Owing to the high velocity of the mixture of air and water in the rising-pipe and the frequent employment of comparatively small pipes for air-lifts, the correct determination of the frictional resistance becomes a matter of great importance. The practice of putting the air-pipe in the centre of the water-pipe, which for pipes of large diameters has very much to recommend it, becomes for small pipes of considerable length one of the most frequent reasons for low efficiency of the air-lift pump. It should be remembered that the frictional resistance in a pipe does not vary as the area of the pipe, but as the area divided by the circumference, which for cylindrical pipes is equivalent to one-fourth of the diameter in feet. If, therefore, as in the Tunbridge Wells air-lift, the rising-pipe is 7 ins. diameter and the central air-pipe is $2\frac{1}{2}$ ins. inside and 3 ins. outside diameter, the frictional resistance is proportional to the difference of the diameters, and not to the difference of the areas. Although, therefore, the free area of the pipe is equivalent to a diameter of $6\frac{3}{8}$ ins., and the velocity of flow is inversely proportional to this area, the frictional resistance is equal to that of a pipe (7 ins. − 3 ins. =) 4 ins. diameter, and, therefore, much higher than would be the case if the air-pipe were arranged outside the water-pipe.

Although the frictional resistance varies as the specific gravity of the fluid, the formulæ used for finding this resistance remains the same as those generally used for water, because the value of f is expressed in feet-head of the mixture itself, and the decrease in weight corresponds exactly to the decrease in specific gravity.

As the exact velocity of the mixture can only be determined by making a few trial calculations, Diagram No. 1 has been prepared for facilitating the somewhat tedious calculations necessary for finding the value of f. It is based on a simplified expression of Kutter's formula which with a coefficient of roughness $n = 0{\cdot}0115$, becomes very nearly

$$f = l \times \left(\frac{\sqrt{D}+1}{100D}\right)^2 \times v^2$$

where—

f = frictional resistance for, and
l = length of pipe in feet.
D = diameter of pipe in feet.
v = velocity of flow in the pipe in feet per second of the average volume $(Q + V_m)$ of water and air.

The table gives the value of f, for one foot-length of pipe in decimals of a foot, which corresponds to the values given in line 20 of Table I. The actual frictional resistance for the whole length of pipe is given in line 21 of this table.

The diagram shows clearly the enormous increase in friction which may be expected for small pipes, more especially when these pipes are long. Under such circumstances the frictional resistance may be very much more than the actual lift, and no very high efficiency can thus be expected.

Depth of Immersion (h_1).

No hard-and-fast rule can be given on this point, but it may be stated generally that a deep immersion is more economical than a shallow one. A deep immersion, however, generally involves a long length of pipe, and where this pipe is of a comparatively small diameter the increase in frictional resistance may counterbalance the theoretical gain. In many cases where air-lifts are installed compressed air is used for other purposes, and it may be more

economical to use the air at the available pressure either by giving a greater or less depth of immersion, than to reduce the pressure by throttling or to put down an independent air-compressor in order to retain the proportion $h_1 = 2h_2$ which is very often adopted. It is in all such cases best to calculate from the formulæ and data given above what will be the effect of varying the immersion and the air-pressure.

Figs. 4, 5, and 6 illustrate a case where compressed air at a gauge pressure of 27 lbs. per sq. in. was available, but where the water to be lifted was only 8 ft. below the surface, and enters a brick chamber through a stoneware pipe. In this case the proper depth of immersion is obtained by driving a 12-in. pipe to a depth of 53 ft. below the bottom of the chamber, and by plugging the end of it with concrete, so as to form a reservoir for the rising-pipe of the air-lift.

Compressed-Air Pipe.

Although the central position of the air-pipe undoubtedly increases the frictional resistance very materially, it cannot, on the other hand, be denied that it has many important advantages. In an ordinary artesian bore hole it is generally quite impossible to predict what quantity of water may be extracted by means of an air-lift, and to what point the water level may sink when the maximum amount of water is raised. If the air-pipe is in the centre of the bore-hole the whole extra cost of the air-lift consists of this pipe with its nozzle, and the depth of immersion can be regulated merely by lengthening or shortening this pipe. In cases where the water-bearing strata contain running sand, this may be cleared away by letting compressed air escape temporarily below the iron lining of the bore-hole, so as to form a cavity round its lower end where the velocity of the water may be too low to move the sand. Where the rising-pipe is large there is a tendency to form a vortex in the centre of the pipe where air may escape without doing work, and the central position of the air-pipe will prevent this.

The diameter of the air-pipe should be adjusted according to the quantity of compressed air required. The ratio $\frac{V_1}{a}$, where V_1 = volume of compressed air in cub. ft. per sec., and a = area of the air-pipe in sq. ft., should not as a rule exceed 20 ft. per sec., which gives a very moderate amount of friction, which, for a $2\frac{1}{2}$-in. pipe 1,000 ft. long, will not exceed 1 per cent. of the initial absolute pressure, and which therefore in most cases may be neglected altogether in the calculations.

Useful Effect of Air-Lift Pump.

In order to make the results given in line 24 of Table I. directly comparable, the i.h.p. required for compressing the quantity of air found experimentally to the observed pressure has been calculated from the formulæ, firstly, for single-stage compressors

$$I.H.P. = V_0 \times 24{\cdot}1 \left[\left(\frac{p_1}{p_0} \right)^{0{\cdot}2} - 1 \right]$$

and, secondly, for two-stage compressors

$$I.H.P. = V_0 \times 48{\cdot}2 \left[\left(\frac{p_1}{p_0} \right)^{0{\cdot}1} - 1 \right]$$

These formulæ give results agreeing very closely with actual practice, and they allow a mechanical efficiency of 80 per cent. in the air-compressing engine, which is a very fair allowance; although, for instance, the air-compressors at Tunbridge Wells, according to the data given by Mr. W. H. Maxwell, have only given 72 per cent. efficiency. This, however, may have been due to temporary causes, and it would serve no useful purpose to introduce such discrepancies into a comparative statement.

These useful effects should not, however, be considered final. When once the principles of the air-lift pump are thoroughly understood and they are designed according to the simple rules given above, improvements may be confidently expected and far higher efficiencies may be looked for, more especially for deep air-lifts. For shallow air-lifts the maximum efficiency has been very nearly obtained by Messrs. Hughes & Lancaster with their air-lift at Rotherham, of which particulars are given in column 10 of Table I.

Notes on Locomotive and Railway Engineering.

By CHARLES S. LAKE.

Power-Reversing Gear for Locomotives.

THE writer has received from Mr. J. Holden, M. Inst. C. E., locomotive superintendent of the Great Eastern Railway, particulars and drawings of an improved power-reversing gear for locomotives which he has designed, and which is being applied to the most recent engines turned out of Stratford Works.

A locomotive fitted with this gear can be reversed in the usual way by means of the hand-wheel *A*, or it can be reversed by compressed air, the gear for this purpose being operated by the handle *B*. Either of these methods can be utilised at will, as the act of using one system of reversing automatically renders the other system inoperative. The reversing-shaft *C* is connected to the reversing-rod *D* by the arm *E* and to the piston-rod *F* by the arm *G*, a guide *H* being provided for the end of the piston-rod.

To operate gear by hand in the usual way, wheel *A* and screw *I* are rotated; the half-nut *J*, and with it the reversing-rod *D* and arm *E*, are moved one way or the other, the air (compressed or otherwise) in the cylinder *K* being displaced through the valve *L* in a manner hereinafter described. To operate gear by power, handle *B* is moved one way or the other, according to whether engine is required to be put in fore- or back-gear. Whichever way handle is moved, the cam *M* attached to it lifts half-nut *J* out of gear with screw *I*; at the same time the arm *N*, which is also attached to handle *B* by the shaft *O*, causes the valve *L* to rotate, thereby opening one end of the reversing cylinder *K*

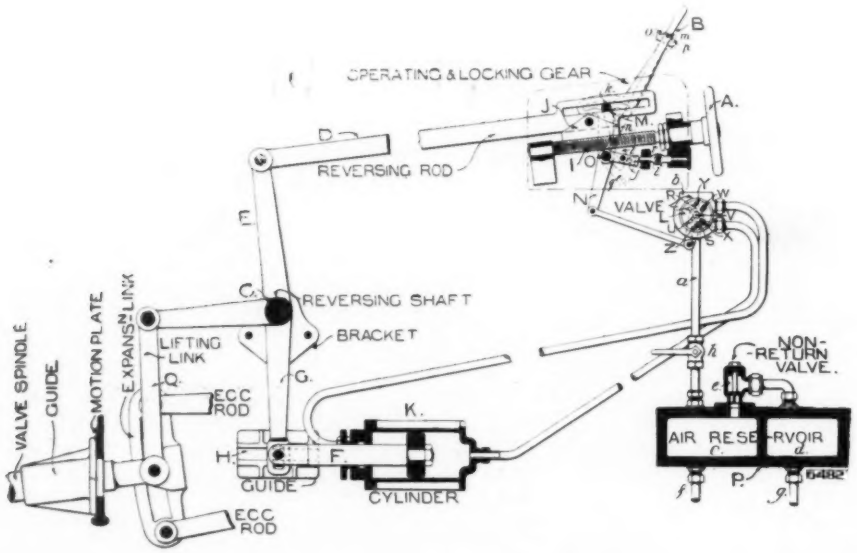


DIAGRAM OF LOCOMOTIVE POWER-REVERSING GEAR.

to pressure and the other end to exhaust. It should be noted that the half-nut *J* is quite clear of screw *I* before this opening of cylinder *K* to pressure and exhaust takes place.

In the running position both ends of the cylinder *K* are in communication with the main air reservoir *P* through the valve *L*, the piston-rod *F* being made of such a diameter that the reduced piston-area, exposed to pressure on the piston-rod side of the piston, balances the weight of the motion hanging on to the lifting links *Q*. The valve *L* rotates on a seating having two ports, *R* and *S*, the port *R* being in communication with the front end of cylinder *K* and the port *S* being in communication with the back end of cylinder. *T* and *U* are exhaust ports cut in the valve *L*; they both communicate with the atmosphere through the hole *V* in the spindle of the valve *L*. *W* and *Y* are pressure ports cut right through the valve *L*. *Y* and *Z* are two holes cut right through the valve *L*, and serve the purpose, when gear is in running position (as drawn), of placing both ends of cylinder *K* in communication with each other through the ports *R* and *S* and with the air reservoir *P* through the pipe *a*. The pipe *a* enters the body of the casting *b* at the back of the valve *L*—viz. on the opposite side of the valve *L* to that where are the ports *R* and *S*.

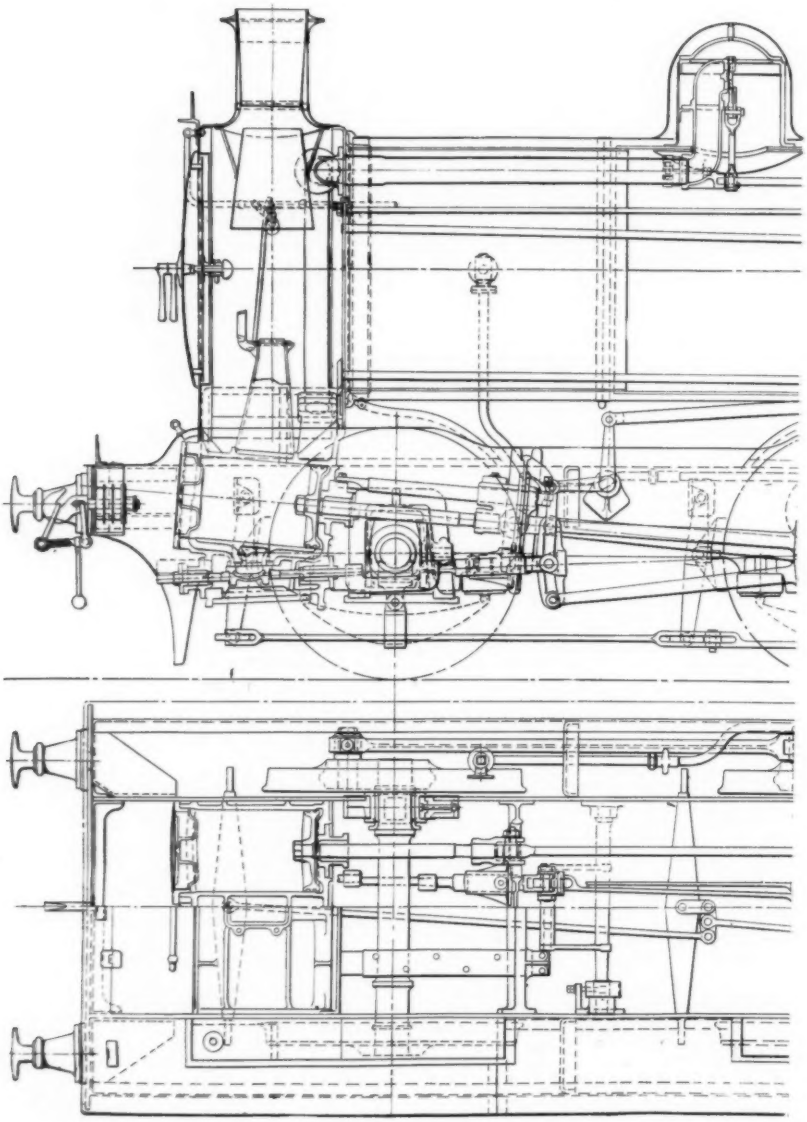
The air reservoir *P* is divided into two parts, *c* and *d*, which communicate with each other through the non-return valve *e*. Air is supplied from the Westinghouse brake pump through the pipe *f* to part *c* of the reservoir, then through the valve *e* to part *d*. The brakes are in communication with the pipe *g*, consequently they can draw air from part *d*, and also from part *c*, through the non-return valve (a considerable quantity of air being required to release the brakes, etc., on a long train). Air for use with the reversing gear is drawn from part *c* only; the valve *e* prevents any air being taken from the side *d* for this purpose. All chance of the unintentional operation of the brakes, due to a reduction of pressure in the reservoir, is thus avoided. The stop-cock *h* is to shut air from reversing gear altogether, should it be at any time necessary to do so. To prevent the possibility of half-nut *J* jumping the screw *I* a locking gear is provided. The pawl *i* engages the lever *j*, the lever *j* being loose on the shaft *O* and connected to the pin *k* by the link *l*. When handle *B* is moved either way it rotates slightly on the pin *m* before moving the cam *M*, and lifts link *n* by means of one of the pins *o* or *p* according to which way handle *B* is moved. The link *n*, in its turn, rotates bell-crank lever *q* and thereby, by means of link *r*, throws pawl *i* out of gear.

Great Eastern Goods Engines.

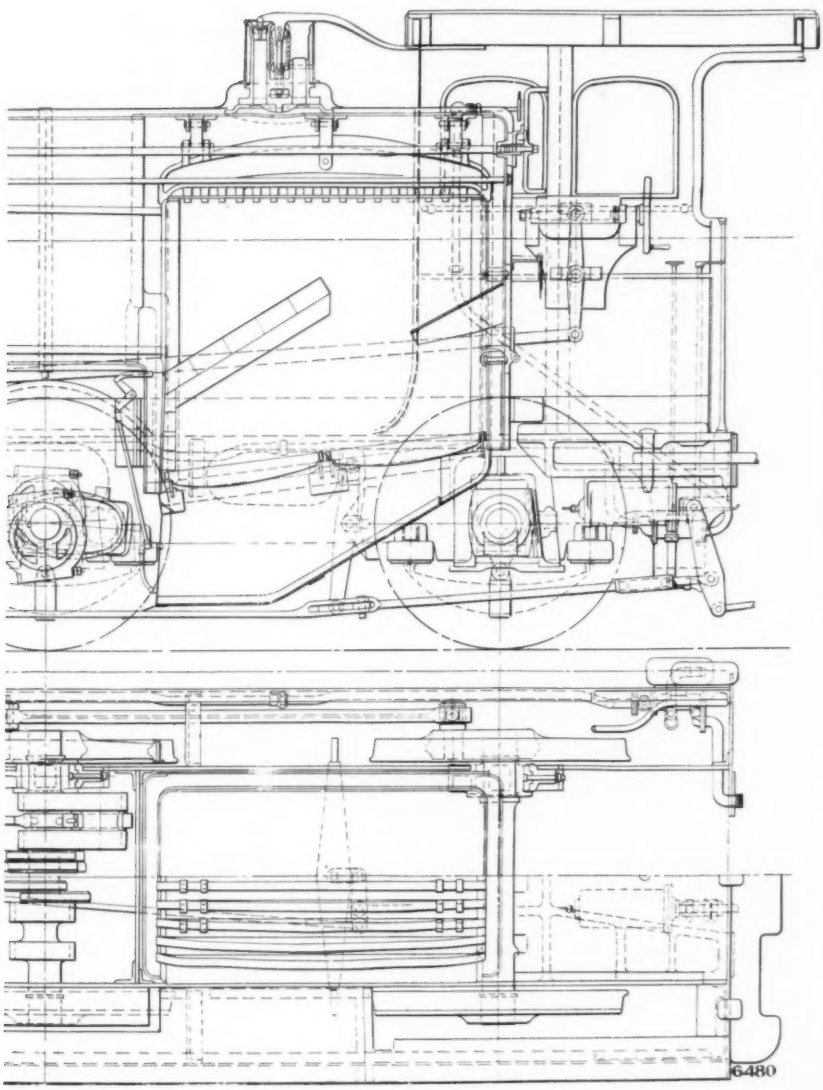
Among the many locomotive designs which Mr. Holden has introduced upon the Great Eastern Railway, the six-coupled goods engines, of which a sectional view appears herewith, are among the most successful. They are engaged in working heavy goods traffic between Temple Mills (the company's vast headquarters of the goods department) and every part of the system. The traffic is of a very considerable nature, and some of the trains are made up to maximum weights; whilst lighter ones, coming within the "fast goods" category, are worked through long distances at high average speeds for this class of service. The locomotives referred to are of the standard pattern on British railways, having two inside cylinders and six coupled wheels, without supplementary carrying-wheels either in front or to the rear of the coupled wheels. The valves, which, it will be noticed, are of the balanced flat type, work below the cylinders at an angle opposite to that of the cylinders themselves; this, of course, being unavoidable where the underneath position is resorted to. The advantage secured—viz., that when steam is shut off the valve falls from the face of the ports, thus avoiding friction—is considered by many engineers to justify the adoption of this practice, which is largely applied to locomotives on the Midland, London, Brighton & South Coast, and other railways. It will be noticed that the reversing-lever is attached at the cab end to the lower end of a vertical arm, which is secured at the upper extremity to the reversing-screw. By this means the handwheel and screw may be placed in a horizontal instead of an inclined position. The cab provided is of commodious design, affording ample protection from weather conditions, whilst being well ventilated by means of adjustable windows. The six coupled wheels are 4 ft. 11 ins. in diameter on tread, and the cylinders 19 ins. diameter by 26 ins. stroke. The writer understands that the engines have given great satisfaction and that their number will be added to. No. 1189, one of this series, has been fitted with a Belpaire firebox, and Mr. Holden is also adopting this pattern in his latest express engines of the "enlarged" Claud Hamilton class.

New Suburban Carriages for the L. & N.-W. Ry.

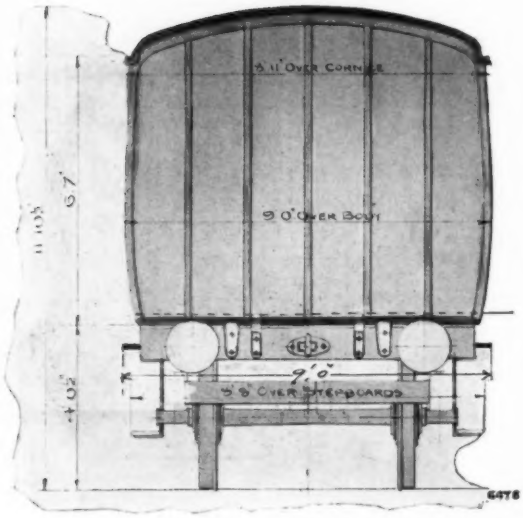
Mr. C. A. Park, M.Inst.C.E., carriage superintendent of the London and North-Western Railway, has during the last year or so introduced a number of new suburban



GREAT EASTERN RAILWAY: NEW SIX



WHEELS COUPLED GOODS ENGINE.



LONDON AND NORTH-WESTERN RAILWAY: NEW SUBURBAN-TRAFFIC CARRIAGE. END VIEW SHOWING OVERHANG AND CAMBER.

carriages upon the railway in question. The local traffic between Broad Street and Watford, and also in the Manchester, Birmingham, and other districts, has been growing very rapidly during recent years; and, whilst it has been considered inadvisable to increase the length of the trains to any great extent, it has been fully recognised that more adequate provision must be made for the ever-increasing number of passengers patronising the company's services in the districts named. Mr. Park therefore decided to introduce new carriages having a higher seating capacity, and there is now running a number of these trains, which the writer has no hesitation in pronouncing the best and most comfortable suburban stock on a British railway. The interior arrangements are well and handsomely appointed, and the exterior of the coaches is fully up to the London and North-Western standard, which leaves nothing to be desired. Each carriage is 50 ft. long by 9 ft. wide, 7 ft. 10 ins. at centre and 6 ft. 7 ins. at the top outside cornice, all outside dimensions of the body. The sides turn in from the 9-ft. waist to 8 ft. 9 in. under cornice to clear the loading-gauge of the line, which is 9 ft. wide, the cornice width being 8 ft. 11 ins. over all; the inside height of the carriage being 7 ft. 6 ins. at centre and 6 ft. 4½ ins. at side to top of cornice.

The carriages are divided into the following compartments:—

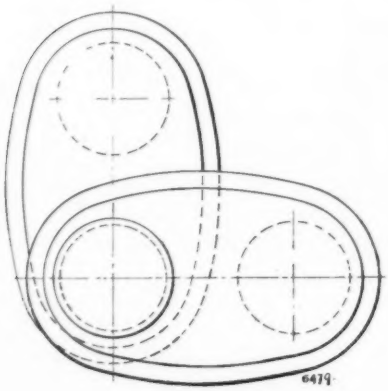
			ft.	ins.	
For first class	... 7	compartments	... 7	0½	length.
,, second class	... 8	,,	... 6	1⅜	,,
,, third class	... 8	,,	... 6	1⅜	,,
Second class, brake end	... 6	,,	... 6	1⅜	,,
Third class, brake end	... 5	,,	... 6	1⅜	,,

the width across being the same for all classes, the seating accommodation for each compartment bejng 8 for first, 10 for second, and 12 for third-class passengers.

Some of the second and third-class brake-end carriages have only five compartments, giving more luggage accommodation, which is required in some districts more than others where such trains are working—increasing the accommodation from 12 ft. 4 ins. to 18 ft. 6 ins. by 8 ft. 4 ins., the width of carriages inside.

The doors at side of carriages are 2 ft. wide for all classes, except for guard and luggage; these are made double, viz., 4 ft. wide.

The structural framing of the body is in teak and oak, the outside being panelled with canvassed and blocked Honduras mahogany.



GREAT EASTERN RAILWAY: STEEL CRANK AXLE, SIX WHEELS COUPLED GOODS ENGINE.

The internal finishing of the compartments for first class is polished American walnut framing filled in with lincrusta panels for back and side quarters. The roof is also covered with bordered lincrusta panels with floral centres, and the whole of the lincrusta is painted enamelled white.

Glazed photographic views and mirror in centre are fixed in walnut frames with bilectioned charrier mouldings above trimming-lace and under light-luggage racks, extending the whole width of the carriage.

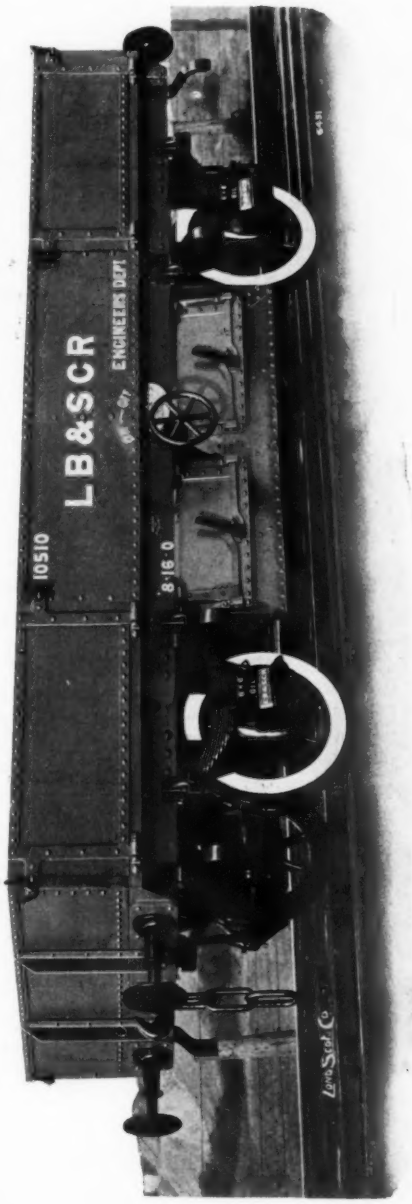
The upholstering of seats and backs is in Saladin moquette with laces to match, and arm-rests covered in leather, except the smoking-compartments; these are upholstered in green embossed buffalo-hides with laces and arm-rests to match also. Blinds are in tapestry, and the floors covered with Kork matting.

The second-class finishing is in polished wainscot oak for backs and side quarters, filled in with lincrusta panels. The roof is also in lincrusta fixed up with mouldings, and painted enamelled white and decorated in the raised parts. Glazed, framed photo-views are fixed above trimming-lace under light-luggage rack. The seats and backs are upholstered in crimson and black velvet, with laces to match. Balanced blinds are in tapestry, and floors covered with inlaid linoleum.

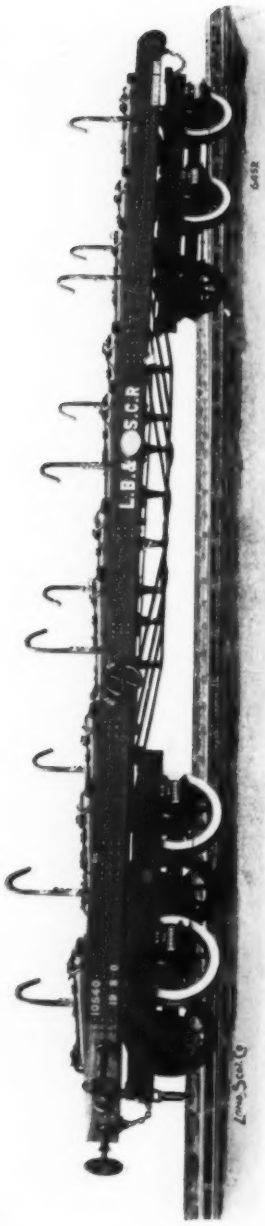
The third-class finishing is only partly in polished wainscot oak, for photographic views and cornice over doors; the other part being painted and grained in oak round doors, with backs painted white. Lincrusta roofs are similar to the second, and decorated. The upholstering is in wool rep brown and gold, and blinds in tapestry, with floors covered in granite linoleum.

The guard's compartment and luggage at each end of the second and third-class brakes have all necessary fittings, viz., shelves, despatch letter-rack, together with pillar hand-brake when needed. The walls are painted light green, with ceilings white.

The underframes for these carriages are all of the same construction, and are made of rolled-steel bulb-angle sides with steel-channel longitudinal and headstocks framed and riveted together, and well trussed with two $1\frac{1}{2}$-in. rods at side and two $1\frac{1}{4}$-in. rods at centre from bogie bolster beams, supporting the two cross transom



STEEL WAGON CONSTRUCTED FOR THE LONDON, BRIGHTON AND SOUTH COAST RAILWAY BY MESSRS. HURST, NELSON AND CO., MOTHERWELL.



BOGIE WAGON CONSTRUCTED FOR THE LONDON, BRIGHTON AND SOUTH COAST RAILWAY BY MESSRS. HURST, NELSON AND CO., MOTHERWELL.

beams, which carry the sides and longitudinals and keep the centre from sagging when loaded with brake-cylinders and accumulator-cells. These underframes are mounted on the standard four-wheeled bogies with 8-ft. centres.

All the carriages are lighted electrically on Stone's system, having dynamos and storage-cells suspended from underframe and driven from the wheel axle. They are also fitted throughout with the Consolidated steam-heating system.

The total number of passengers carried when a train of five vehicles is used is: 56 first-class, 60 second, and 252 thirds, making a total of 368 passengers.

Another Cylinder Arrangement for Compound Locomotives.

The American engineering press is giving prominence just now to a new compound locomotive, which has been built by the American Locomotive Company at their Schenectady Works for the New York Central and Hudson River Railroad. In this engine, which is of the "Atlantic" type, the low-pressure cylinders occupy the customary position outside the frames and drive the second pair of coupled wheels; whilst the high-pressure cylinders are carried out in advance of the smokebox and drive the cranked axle of the first pair of coupled wheels. Commenting upon the design, one of the principal engineering journals published in the United States remarks:—

"The locomotive represents a careful effort to provide as many of the essential elements of the famous De Glehn four-cylinder compound as appeared to the designer advisable to introduce into an American locomotive at this time. The important elements are four cylinders for compound working, so arranged that their forces, which constitute a disturbing element in the ordinary construction of locomotives, oppose and neutralise each other to produce a balanced engine. The work is divided among four cylinders in such a way that each part has less duty to perform than in the usual construction; and, furthermore, this engine is arranged to divide the cylinder stresses between two driving-axles, instead of concentrating them all upon the crank-axle. The introduction of this engine marks further evidence of faith in the principles so successfully applied in the De Glehn compounds in Europe. In this design only one valve-motion is employed on each side of the engine, and thus stops short of De Glehn's complete idea. The advent of a balanced and divided compound by the American Locomotive Company, together with the ready acceptance of four-cylinder balanced compounds built by the Baldwin Locomotive Works, certainly seems significant of a new era in American locomotive design."

So far as wheel arrangement is concerned the engine differs nothing from others of similar type, and at first sight it has the appearance of being an ordinary two-cylinder 4-4-2 locomotive. A second glance, however, discloses the presence of the high-pressure cylinders ahead of the smokebox, but it cannot in any way be said that this detracts from the general appearance; indeed, the knowledge that the engine is of a later and more advanced description is sufficient to overcome any opinion one may form with regard to appearance, which is, after all, a secondary consideration. The high-pressure cylinders are 15½ ins. diameter by 26 ins. stroke; low-pressure cylinders 26 ins. diameter by 26 ins. stroke; coupled wheels 6 ft. 7 ins. diameter on tread; boiler: diameter 6 ft. 0¼ ins., length of tubes 16 ft., number of tubes 390—2 ins. diameter, heating surface (total) 3,446·1 sq. ft., and working steam pressure 220 lbs. per sq. in.

New Wagons for the L.B. & S.C. Railway.

THE accompanying illustrations show two types of wagons recently constructed by Messrs. Hurst Nelson & Co., at their works at Motherwell, N.B., for the engineers' department of the London, Brighton & South Coast Railway.

The wagons are built entirely of steel, and, as will be noticed, they are fitted with oil axle-boxes. The four-wheeled wagon is to be used for ballast, and is intended to carry a load of 20 tons. The bogie wagon is to be employed for the conveyance of rails and sleepers. On one side of this wagon the removable side stanchions are shown when in position for unloading the rails, and this constitutes a special feature of the wagons. Ten mild steel bolsters are rivetted to the wagons, and these are specially pressed by the builders in their large hydraulic press. The usual binding chains with screw couplings, etc., are supplied for securing the load, and altogether the wagons present every indication of being very serviceable vehicles.

They were built in accordance with the specifications and drawings prepared by Mr. R. J. Billinton, Locomotive and Carriage Superintendent of the London, Brighton & South Coast Railway, and the work was carried out under the supervision of Mr. L. B. Billinton. They represent a genuine advance upon previous practice in this class of wagon.

Internal Combustion Motors.

By DUGALD CLERK, M.Inst.C.E.

(Continued from Vol. 10, page 506.)

TAKING another deviation to imitate what sometimes occurs in practice, the heat addition is assumed to be made so that one portion is added at constant minimum volume, another part is added at constant pressure during the first tenth of the stroke of the piston, then the expansion is continued $PV^{1\cdot2}$ to the end of the stroke. Compression is still $PV^{1\cdot408}$. In this case, with maximum temperature 1,600 degs. Cent., and suction temperature 0 deg. Cent., E_1 is 0·406 ; while with the same maximum, but suction temperature 100 degs. Cent., E_2 is 0·358 for $\frac{1}{r}$ = one-tenth. Here there is a great falling off from 0·61 to 0·406 and 0·358 depending on suction temperature. These, however, are extreme deviations, which are avoided as much as possible by gas-engine designers. It is well known to gas-engine constructors, that to get the most economical diagram possible the ignition must be fairly sharp and the maximum temperature must come as nearly as possible to the compression end of the stroke. Igniting is always adjusted in actual engines to produce this condition. It is not very common to have expansion curves so flat as $PV^{1\cdot2}$, but expansion $PV^{1\cdot3}$ is very common. Taking here compression as $PV^{1\cdot408}$ and $\frac{1}{r}$ equal to one-fifth, efficiencies are as follows :—

1,600 degs. Cent. to 0 deg. Cent. $E_5 = 0\cdot45$

1,600 degs. Cent. to 100 deg. Cent. $E_6 = 0\cdot446$

This case corresponds to a more common expansion curve, and here it will be seen that the number 0·45 or 0·446 deviates but little from the standard efficiency, which is 0·47.

In all these cases the expansion is not considered as adiabatic. It has been assumed that the specific heat of air is constant. If, however, it be ultimately proved that the specific heat of air is not constant, the efficiency numbers obtained from similar calculations will not be greatly changed, because variable specific heat assumes a change in the internal energy of the working fluid, which is equivalent in its effects to heat added during an expansion stroke ; that is, the heat which has to appear later in keeping up the expansion line above the ordinary adiabatic assumed for air must be stored up in the working fluid, in the first instance, because of the increased specific heat at constant volume. Whichever proves to be true, constant specific heat with continued combustion, or varying specific heat with completed combustion, the effect upon efficiency will be very much of the same order as has been here calculated.

Notwithstanding the knowledge that our air cycle will deviate from a cycle of constant efficiency under the assumed conditions, yet the gas-engine cycle may be considered to be fundamentally one of this nature. In all modern engines of the explosion type, the attempt is made to add all the heat as early in the stroke as possible, because heat added at the beginning is generally recognised as being more effective than heat added towards the end. This becomes evident on comparing the efficiencies obtained in actual engines with the standard efficiencies given at Table II. Table I. shows, in the column headed "E, standard air cycle," the efficiencies calculated in accordance with Table II. ; that is, the efficiencies of the standard air cycle for the particular compression volume of each engine, the compression volume $\frac{1}{r}$ is given in the next column. In the column $\frac{\text{"I.H.P."}}{E}$ the ratio of the actual efficiency to the standard efficiency is given, and there it will be seen that this varies from 0·48 in Slaby's experiment to 0·6 in Robinson's. The ratio is practically the same in the Humphrey test of the large Otto engine, and in the Robinson test of a National engine.

This comparision of efficiency of a standard air engine with that of the actual engine is even more interesting in cases where a series of experiments have been conducted with the same engine, varying compression only. Such experiments have been made by Professor Meyer in Germany, and by Professor Burstall in England. Taking Professor Meyer's results, I have calculated out the air standard efficiency for the different compressions, which ranged from 40 lbs. per sq. in. to 80 lbs. per sq. in. above atmosphere.

In Table V. the results of this comparision are shown. The first column shows the actual efficiency, the second the air standard efficiency, and the third the actual upon standard. The fourth column shows the revolutions per minute of the engine.

Table V.—Calculated from Professor Meyer's Experiments.

Actual efficiency.	Air standard efficiency.	Actual / Standard.	Revolutions per minute.	Dimensions of engine, etc.
0·250	0·44	0·58	257	Engine 7·8-in. diameter by 11·8-in. stroke. Compression varies from 40 lbs. to 80 lbs. above atmosphere.
0·244	0·42	0·58	249	
0·214	0·37	0·58	251	
0·188	0·33	0·57	225	

In this case, although the air standard efficiency varied from 0·33 to 0·44, the ratio of the actual efficiency to the standard was practically constant at 0·58. In the case where the number becomes 0·57, this is evidently due to the lower rate of revolution of the engine in that test.

The first and second parts of Table VI. show the actual and air standard efficiencies calculated by me from Professor Burstall's test—Gas Engine Research, Institution of Mechanical Engineers—for the highest and lowest results.

Table VI.—Calculated from Professor Burstall's Tests.

	Actual efficiency.	Air standard efficiency.	Actual / Air standard.	Maximum temperature.	Dimensions of engine, etc.
		First.			
A^7	0·189	0·33	0·57	1145°C.	
B^9	0·212	0·30	0·59	1098°C.	
C^7	0·219	0·43	0·51	1154°C.	
D^8	0·231	0·47	0·50	1094°C.	6-in. diameter by 12-in. stroke. Revolutions about 200. Compression from 30 lbs. to 105 lbs.
		Second.			
A^1	0·166	0·33	0·50	1751°C.	
B^2	0·187	0·36	0·52	1745°C.	
C^1	0·172	0·43	0·40	1749°C.	
D^1	0·181	0·47	0·38	1437°C.	

The first and last columns have been taken from Professor Burstall's tables ; the second and third columns have been calculated from the details given by him.

It will be observed that all the higher efficiencies were obtained with maximum temperatures of about 1,100 degs. Cent., and all the lower efficiencies with temperatures of about 1,700 degs. Cent. It will be further observed that in the first two tests, A^7 and B^9, the ratio is 0·57 and 0·59. In the second two, C^7 and D^8, it falls to 0·51 and 0·5. In A^1 and B^2 also this ratio is less—0·5 and 0·52 ; but in C^1 and D^1 it falls to 0·4 and 0·38.

These experiments very clearly show the prejudicial effect of high temperatures in increasing heat losses, and also the evil effect of cooling surfaces of certain configurations.

From these numbers it may be taken that in motors of ordinary commercial constructions of to-day the actual efficiencies vary from 0·5 to 0·6 of the air standard efficiencies, depending on the dimensions of the engine and the success of the designer in arranging the shape of the combustion space ; and also, of course, upon the perfection of the action of the valves. The deviation of the actual from the air standard efficiencies is, of course, due to heat losses to cylinder and piston, heat additions at improper periods, varying specific heats due to the fact that the working fluid of the gas engine is not pure air, and possibly to change of specific heat with changing temperature.

Many useful lessons can be gathered from these tables. From them it is evident that, under whatever conditions the assumed air engine be operated, diminution of compression volume means increased efficiency ; and from the comparisons of the actual with the

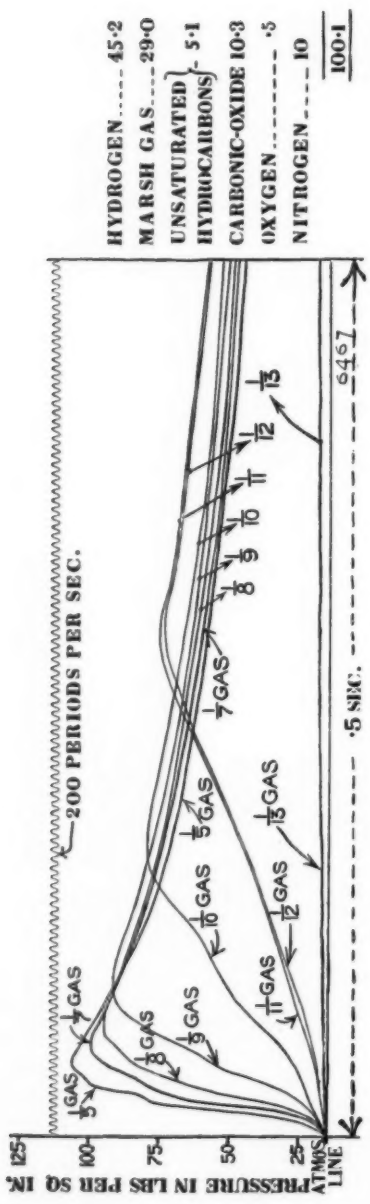


FIG. 1.—EXPLOSION AND COOLING CURVES FOR VARIOUS MIXTURES OF LONDON COAL GAS.

air standard efficiencies it is also apparent that large engines approach more nearly to the standard efficiencies than small ones, and that loss is occasioned by high-flame temperature and by improperly disposed cooling surfaces.

Although it is evident that the practical results follow the standard results with some closeness, yet much information is required before a really accurate standard engine of comparison could be formulated. Before this can be done considerable investigation is required into the actual conditions found in gaseous explosions, and perhaps the first question which must be definitely settled is that of the specific heat of air and the various gases which enter into the chemical action of combustion at temperatures ranging from, say, about 500 degs. Cent. to about 2,000 degs. Cent. No satisfactory determinations have been made at these high temperatures. It is true that four distinguished French physicists, Messrs. Mallard, Le Chatelier, Berthelot, and Vieille, have attempted to obtain values of these specific heats by following the beautiful method of investigation originally used by Bunsen. They took various mixtures of inflammable gases with oxygen and air, added diluted gases, such as nitrogen, carbonic acid, steam, etc., and determined the various pressures and temperatures produced by gaseous explosions in these varied proportions. From these experiments they attempted to deduce the specific heat of oxygen, nitrogen, carbonic acid, steam, and other gases. To enable them to do this, they knew, in the first instance, the amount of heat in the shape of inflammable gas present for each explosion. They satisfied themselves that combustion was complete at maximum temperature, that dissociation was absent, and that other disturbing causes were eliminated. They then came to the conclusion that the specific heat of these gases had greatly increased at high temperatures. In my opinion these conclusions are vitiated by what I consider to be the fundamental fallacy of supposing that any mathematical examination of lines of cooling could determine with any accuracy whether combustion was completed or not. I have made many experiments on this subject, and my conclusion is that combustion is not complete when those distinguished experimenters assumed it to be so. All gas-engine explosion experience is against the assumption of any possibility of proving changing specific heat by combustion experiments. I have made a series of experiments upon London coal-gas with a new apparatus of great accuracy. The curves produced are shown at Fig. 1. The analysis of the gas is marked on the figure. From these cooling curves I have selected a

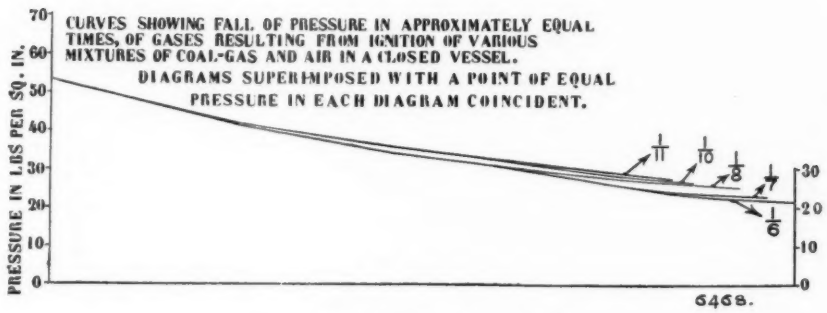


FIG. 2.—FALL OF PRESSURE IN EQUAL TIMES.

common pressure, and measured carefully the rate of cooling from that common pressure. If the specific heat be changed, then the rate of cooling will be constant from any given temperature. The choice of a common pressure means that at the particular period of time in the explosion vessel the mean temperature of the gases is the same. If the temperature of the gases be the same, and the cooling surfaces be of the same temperature, then the falling curve for all mixtures should be invariable.

Fig. 2 shows the falling curves superimposed the one upon the other from the same point. It will be found that they do not cool at equal rates. The rates are not greatly different, but the difference is perfectly easily observed, and it follows a rigid general law. The richer the mixture exploded in inflammable gas the more rapidly does it cool from the same temperature; and it will be observed that in mixtures varying from 1 in 6 to 1 in 11 the cooling curves are ranged the one upon the other in reverse order of the richness of the mixture. This experiment is quite inconsistent with the idea that the only phenomenon going on in the vessel at these periods is change of specific heat. Obviously in these experiments heat is being added in each case—in some cases at a greater rate than in others.

The phenomenon familiar to all gas engineers of a high-expansion line in cases of weak mixtures also proves that some method of adding heat is present other than any mere change of specific heat. By properly proportioning the mixture in a gas-engine cylinder, even firing it quite completely at the beginning of the stroke, it can be shown that almost any desired curve of fall may be obtained up to the isothermal line; and this, of course, is quite inconsistent with the hypothesis of change of specific heat.

The physicists would do great service to the gas engineers if they would determine the specific heat of oxygen, nitrogen, steam, carbonic oxide, and hydrogen, at these high temperatures by some methods which do not involve combustion. Several methods suggest themselves.

The lecturer then briefly discussed several ways in which this might be done.

The determination of the maximum temperatures of gaseous explosions depends, to a large extent, on a knowledge of the chemical actions going on in an explosion. These actions are of a very complicated kind, and more exact knowledge as to their nature will exercise considerable influence upon the future of the internal-combustion motor. Assuming complete combustion, it is well known that a maximum explosive mixture of hydrogen and oxygen, or carbonic oxide and oxygen, suffers a contraction of one-third in the passage from the uncombined gases to the compounds, steam or carbonic acid. In the use of hydrocarbons such as occur in petroleum and petrol—hydrocarbons, say, such as butylene, amylene, pentane, heptane, and octane—it is found that, instead of contraction, molecular expansions occur.

With alcohol, for example, which is now coming into use in these engines, eight volumes of explosive mixture expand to ten volumes on complete combustion. An air mixture with alcohol could easily give an expansion of about 6 per cent. In this respect alone, then, all change of molecular volume between the uncombined gases and the compounds formed after combustion must be carefully examined before the temperature can be estimated, either from an indicator diagram taken from an engine or from the results of an explosion in a closed vessel.

It is absolutely necessary, however, before any accurate standard of comparison can be formulated, to know with some exactness the

real actions occurring in gaseous explosions, as bearing upon the proportion of combustion at given stages of the explosions.

A further question has to be considered in this regard in the internal-combustion motor, the cycle of operations, charging and compressing, proceeds with one set of chemical substances, and the combustion at once changes those substances into other compounds. The question then arises in calculating the heat added at constant volume, which specific heat is to be taken, the mean specific heat of the working fluid before combustion, or its mean specific heat after combustion. This question cannot be definitely answered without knowing the exact history of the change of the chemical nature of the working fluid; so that, in calculating from existing engine diagrams, it is impossible to say at present with accuracy even how much heat has been added at the minimum volume, and how much is added during expansion. That heat is added during expansion in many cases does not seem open to doubt; but how much heat is added depends upon a knowledge of the chemical history of the working fluid within a period, in a large engine, of about one-sixth of a second, and in a small motor-car petrol-engine within a period of one-twentieth to one-fiftieth of a second. These are the facts which the engineer requires before the properties of an engine can be formulated to be used as a really accurate standard engine of comparison.

The lessons of the simplified standard engine of comparison which has just been discussed are very clear, and gas-engine constructors have noted that among all uncertainties there emerges one certain fact, that the higher the compression — or, rather, the smaller the compression space relatively to the cylinder volume—the greater the economy to be obtained from the engine. They have seized upon this fact, and the past twenty-five years of English and Continental practice have shown steadily increasing compression. Improvement in this direction may be considered as due to bettering the conditions of the thermo-dynamic cycle; but another line of improvement is also open, and that line is to reduce the heat losses to the lowest possible point. All internal-combustion motors suffer large heat losses because of the contradictory requirements of the practical cycle. The cycle requires a charge of working fluid—first as cold as possible, then the compression of that charge without heat loss or gain, then combustion producing high temperatures, succeeded by expansion, also without heat loss or gain. In an ordinary internal-combustion motor of medium size these opposite conditions follow each other in the same cylinder at intervals of about one-sixth of a second, and in a high-speed petrol-motor at from one-twentieth to one-fiftieth of a second. It is, therefore, impossible to arrange cylinder conditions to minimise heat flow, because any attempt to keep surfaces hot to prevent heat loss at once introduces heat into the working fluid at a time when it should remain cold. Some heat loss, therefore, from the hot gas to the relatively cold walls is unavoidable. This loss, however, becomes naturally less and less with increase in the dimensions of the engine.

With engines of similar proportions the surface exposed for cooling flame increases as the square, while the capacity of the engine increases as the cube. From this it follows that the larger engines should lose less heat proportionately than the small ones. This is found to be the case, but it is also found that in engines above certain dimensions the reduction of cooling surface can be pushed too far. This is due to the fact that an engine piston has to compress an inflammable mixture to high pressure before ignition occurs at all. If the products of the previous combustion be not entirely removed, then in a large engine the temperature remains so elevated that the initial or suction temperature of the engine is raised, and at high compressions pre-ignitions occur.

This question of pre-ignitions limits the engine designer in many ways, and it becomes absolutely necessary to preserve the inflammable mixture during compression at a temperature below its igniting-point. Any inflammable mixture only requires to be sufficiently compressed to ignite without any other means of inflammation. Unfortunately, the conditions of maximum economy in engines are also conditions which favour pre-ignition. In small engines pre-ignitions are rare and are not dangerous; in large engines they are, unfortunately, somewhat frequent, unless mean pressures be kept down and extensive cooling adopted, and when they do occur they are somewhat dangerous.

At present the engineer knows, from previous experience, that if he carries his compression beyond a certain limit for a certain gas he will at once make his engine relatively unworkable by frequent pre-ignitions. The great limiting element in all internal-combustion motors, so far as high compressions are concerned, is this one of pre-ignition. Pre-ignitions are due to many causes—sometimes to a small incandescent part of metal within the cylinder, sometimes to a porous surface of metal within the cylinder, sometimes to a hot piston-end, sometimes to an overheated exhaust-valve, possibly to charred matter of carbon deposited in the cylinder. In all cases, however, pre-ignition occurs owing to

the tendency of these engines to be overheated. In large engines of perhaps 200 h.p. and upwards pre-ignitions are only avoided by excessive water-cooling of all parts and a plentiful flow of cooling water at low temperature.

Definite chemical information is much wanted on the ignition points of various gaseous mixtures when treated under internal-combustion motor conditions in fairly large volumes. Experiments such as chemists sometimes conduct, in narrow glass or metal tubes, are of little use in considering this question of pre-ignition. Such experiments, if made, would require to be performed with comparatively large bodies of material exposed for definite fractions of a second to certain graduated temperatures.

It has been found, then, that in large engines the reduction of cooling surface can be carried too far ; in small motors the better the surface arrangements the better for the economy of the engine ; but in large engines the practical point is the more important point, so that cooling surfaces are deliberately increased in proportions for the purpose of reducing this difficulty.

While on this point, it must never be forgotten that the engineer frequently deliberately abandons the path of theoretical rectitude in order to obtain success in the practical operation of his mechanism. It by no means follows that he is ignorant of this path, as is too often assumed by some of our more theoretical friends.

It will therefore be seen that the temperature at which pre-ignition occurs depends on the nature of the gas used ; and so the engineer may regulate his compression, rate of cooling, and other circumstances, to keep his mixture lower than this point. From this it will be seen that one of the great limitations imposed on the large engine is due to excessive heat-flow through the enclosing walls. It is well known to engineers that this heat-flow increases with increase of explosion temperature. Notwithstanding this, in the smaller engines the tendency of recent years has been to increase flame temperatures within the cylinders to the highest safe economical point, because each maker desires to sell an engine giving the greatest power as well as economy for a given price. The problem is, in the end, a commercial one, and the pressure of competition in the end forces the designer to get not only the highest economy, but also the highest safe-power limit for compressions and dimensions of the engine.

This question of high mean pressure is a most important one. In many of the large gas engines the mean working pressures are relatively low ; in the large continental engines mostly 60 lbs. per sq. in. or under. British-built engines usually operate the medium sizes at pressures as high as 90 lbs. and 100 lbs. per sq. in ; and the English engineer at present is by no means satisfied with the average pressures which he can safely employ in existing large gas engines. To obtain high mean pressures is to have an engine of great power for small dimensions ; but if flame temperatures be reduced to prevent heat-flow difficulties and save heat in this way, the mean pressure is also reduced. It is very desirable to reduce heat flow, especially in large gas-engines.

Some time ago it appeared to me possible to reduce maximum temperatures while increasing mean pressures by increasing the charge weight per stroke given to an engine. I had experimented with two engines, one of 7-in. cylinder, 15-in. stroke, and another of 10-in. cylinder by 18-in. stroke. These engines, which are of the ordinary standard four-cycle type, are allowed to take in the usual charge of gas and air ; then at the end of the stroke a further charge of air or other inert fluid is added to increase the pressure in the cylinder to 7 lbs. or 8 lbs. per sq. in. above atmosphere before the return of the piston begins. This has the effect of increasing the charge weight present in the cylinder by about 40 per cent., and of increasing the pressure of compression, without, however, increasing the temperature of compression. Indeed, in both experiments the temperature of compression was diminished. As the charge present is constant so far as gas is concerned, the maximum temperature capable of being produced is much reduced. The maximum temperature in the diagrams taken by me from these two engines is about 1,200 degs. Cent. Experiments were made, and it was found that the heat flow was reduced to about two-thirds ; and, further, that the mean available pressure was increased about 20 per cent.

Fig. 3 shows two indicator diagrams from the 10-in. by 18-in. engine. One diagram is from the engine in its ordinary state, the other diagram shows the effect of adding the air. In both cases the gas present per charge is practically the same. It will be observed that 20 per cent. more power is obtained with a proportionately reduced consumption of gas. The figures obtained have been marked upon the diagrams.

This method supplies another means of increasing the economy of these engines without changing the thermo-dynamic efficiency, simply by reducing heat losses ; that is, in these cases, the economy is effected with a heat cycle of unchanged thermo-

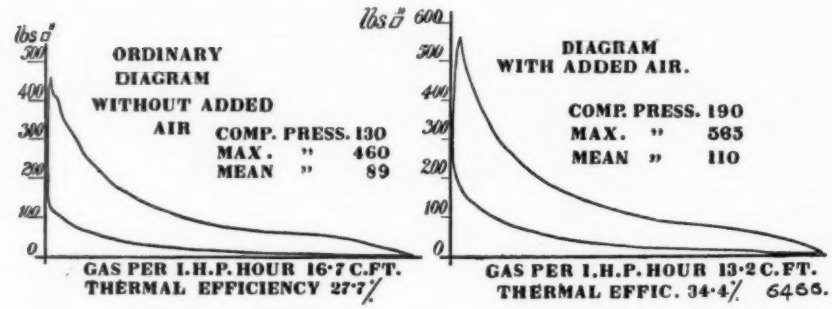


FIG. 3.—AVERAGE DIAGRAMS FROM 10-IN. BY 18-IN. ENGINE.

dynamic efficiency by effecting a change in the heat-flow through the cylinder walls. This method of increasing practical efficiency has the merit of reducing temperatures all round and preventing pre-ignitions in two directions—namely, by temperature reduction, and also by reducing the average inflammability of the charge, so that it is less readily ignited on compression.

The large National gas-engine is designed to work with high mean pressures and low-flame temperatures. The front end of the cylinder is used to compress air into a space. This air is passed through ports to the charge which has been sucked in on the other side, and the charge weight is thus increased without adding additional inflammable material. When compression takes place the pressure rises to about 200 lbs. per sq. in.; then, on ignition, pressure is obtained equal to about 500 lbs. per sq. in. This engine has been designed for mean available pressures of about 130 lbs. per sq. in., and maximum flame temperatures of about 1,200 degs. I cannot but think that it is desirable in large engines to keep cylinder dimensions as small as possible, in order to minimise expansion difficulties, and to raise the maximum working pressures to limits far above those at present used. This method of reducing temperatures appears to be worthy of great attention.

The comparison of the results actually attained in practice with the results calculated for a standard air-engine shows very conclusively why efficiencies have not risen more in proportion to the diminished heat losses through the cylinder walls. The lesson the comparison teaches is, that if in large engines just enough heat could be added to the expanding gases to keep the expansion line up to the adiabatic, then greatly increased efficiencies would be obtained, and no increased loss of heat would be suffered by the discharge of the exhaust. From the expansion curves, however, of large engines it is evident that heat is added during the expansion stroke in proportions far greater than is necessary to keep the line up to the adiabatic. More work is done above the adiabatic line, and this at once introduces a cause of loss of efficiency, as is shown clearly by the tables referred to. Further, it also introduces an increased loss by discharge of a certain proportion of heat with the exhaust gases. This loss, as has been noticed, is increased in large gas engines instead of diminished. My experiments show a still further reduction of heat loss through the cylinder walls, but also a further increase in the heat loss to the exhaust gases. Engineers should endeavour to find means of adding the heat required at an early part of the expansion in order to avoid this difficulty. It may very well be that the rate of combustion is too slow to enable all the required heat to be added early in the stroke. In large gas-engines it certainly appears as if this were the case, and this peculiarity introduces the possibility of improvement in efficiency by the use of lower speeds giving longer periods for the combustion of each charge. If physicists or chemists could discover means of adding heat early in the stroke greater efficiencies would be at once attained. It might be said that improvement in this respect could be had by expanding to a volume greater than that of the original volume of the gaseous charge before compression. This, no doubt, produces some improvement; but, unfortunately, it greatly increases the weight and dimensions of an engine for a given power. The old idea among gas engineers to increase economy was to increase expansion beyond original charge volume. It has been found, however, that this is much more effectively

done by diminishing compression volume. Greater expansion gained in this way is gained under conditions which reduce cooling surface instead of increasing it, and impose no greater strains upon the engine relatively to the volume swept through than in the case of expansion beyond atmosphere.

Enough has now been said upon the motor itself to show the principles upon which the past twenty-five years' advance has been made, and to indicate to some small extent at least what knowledge is required to formulate an accurate standard of comparision, and what line of advance should be followed to reduce temperatures and increase mean pressures.

Before leaving the question of thermodynamics, might I suggest to the physicists that profitable study might be devoted to the question of the laws governing the efficiency of imperfect cycles. The text-books devote a great deal of attention to perfect cycles of operation, such as the Carnot cycle or the perfect regenerator cycle; but little attention is given to the theory of imperfect cycles, which, after all, are the only cycles which can possibly be be carried out into practice. Many engineers are so imbued with the ideas deduced from the Carnot cycle that they find difficulty in accepting the existence of cycles of constant efficiency, although many such cycles exist. Two in particular are now well recognised, the Beau de Rochas and the Joule or Brayton cycle.

The laws of the cooling of gases under various pressures at high temperatures do not seem to be known with any accuracy. To some extent valuable data might be obtained by measuring the maximum pressures and temperatures of gaseous explosions, both with and without initial compression, in large and in small vessels, for simple and complex mixtures of varying dilutions.

The cooling curves, too, are required in closed vessels for all these mixtures. Similar measurements of such explosions should be made for light and heavy hydrocarbons, for alcohol, blast furnace gas, producer gas, coke oven gas, and all the numerous inflammable gases and vapours now pressed into the service of the internal combustion motor. More determinations of this kind are urgently wanted, as the older experiments are now out of date with the change in the composition of towns' gas and the extended use of the other gases.

The internal-combustion motor offers, perhaps, the best instance of international industry which could be imagined. England stands at the head of all nations in the production of the smaller and medium-sized fixed gas-engines. She has produced altogether something approaching 100,000 engines of an average of 20 h.p. This is greatly in excess of the whole Continental output of engines of this type taken together. On the other hand, Continental engineers have given more attention to the large gas-engine—engines of 500 h.p. and above—and the application of blast-furnace gas to the use of the large engine. In this field of internal-combustion motor work Germany excels.

In the petrol motors for motor carriages France appears to take the lead, and in heavy oil-engines England takes the lead. In America engineers are now giving considerable attention to the internal-combustion motor; but so far, for some reason, the Americans have not shown in this subject the inventive and progressive spirit which they usually display. They have, however, taken considerable part in the application of petrol motors to small vessels and boats.

It may be said, however, without exaggeration, that the whole world is now alive to the possibilities of the internal-combustion motor, and that progress will be more and more rapid. These motors have almost fulfilled the expectations of those engineers who—like myself—have devoted a large part of their lives to their study and advancement. They are looking forward now to the completion of the work begun so many years ago, and expect, at no distant date, to find the internal-combustion motor competing with the steam engine—even in its latest form, the steam turbine—on sea as vigorously as it does at present on land.

New Machinery and Appliances.

Experimental Steam and Explosion Engines.

IT is generally conceded that for the transport of heavy loads by road the most suitable means of propulsion is the steam engine; and the remarkable development of the steam lorry, since the removal of legal obstacles, has been rendered possible by the adoption of high-pressures and quick-revolution engines. The limit of improvement in steam-propelled road vehicles is not perhaps represented by the types now common on the highways, and further development must be looked for either in the direction of the steam generator or in the engine itself. In this connection we have had recently brought before our notice an invention relating to steam and gas engines, which, although not yet in the commercial stage, is of sufficient interest to merit some attention. It consists primarily in the manner in which the usual reciprocating action of an engine, which may be either steam or gas driven, is converted into rotary motion; and, instead of employing the usual piston line and crank, the inventor substitutes therefor a cam motion. This cam, which surrounds and forms part of the piston, is fixed at right angles to the axis of the cylinder, and engages by its groove a roller forming part of the flywheel. Hence, with a two-pointed cam, one revolution of the flywheel would correspond to two strokes of the piston.

The cam principle is also adopted for operating the valves, with the result that the construction of the engine is greatly simplified; and providing that the break-test shortly to be made on a new and larger engine shows that the efficiency obtained compares favourably with a quick-revolution engine of the ordinary type, there would be much to be said in favour of the present invention.

Referring to the illustrations, Fig. 1 is a longitudinal section through a simple, double-acting type engine. Fig. 2 is a transverse section on the line 4_1 4 of Fig. 1, while Fig. 3 represents an external view of a completed engine. The cylinder a_1 is closed at each end as shown at a^1, a^1 in which slides the piston b_1 also closed at each end. The piston is formed with a pair of hollow projections b^1, which extend through longitudinal slots a^2 in the cylinder, the projections enclosing the cam groove c, which encircles the piston.

The flywheel d, is cast in one piece with the sleeve or casing d^1, which runs on ball bearings d^2, carried upon extensions of the cylinder. The flywheel is provided with an internal roller d^3, engaging with the cam groove c. Steam is admitted to the cylinder through the inlet passages e, admission being controlled by the inlet valve e^1, and after expansion, exhaust takes place through the valve f^1. The inlet valves are normally pressed upon their seatings by springs e^2, and are lifted at regulated periods by cam surfaces attached to the inside of the casing d^1.

To enable the degree of cut-off to be regulated, each cam-surface is made in two parts, g and g^1 (Fig. 2), the former being fixed and the latter capable of adjustment so that the length of the cam can be varied. How, exactly, the inventor proposes to effect this adjustment from the driver's stand was not stated at the time of our inspection, but some means other than that of loosening the studs g^2 must be found if the invention is to assume any commercial value.

The cams g operate the inlet valves through a pivoted lever g^4 (Fig. 1), and the exhaust valves are operated by the cams f^2, also attached to the casing.

The cam groove is lubricated by a passage d^4 in the flywheel. With a two-pointed cam constructed as shown in the drawing, it will be seen that the fly-wheel makes one revolution to each complete to-and-fro movement of the piston; with a four-pointed cam one revolution of the fly-wheel would need four strokes of the piston; hence the cam motion may be utilised for the purpose of gearing the engine down to a desired ratio, if such were needed.

It is proposed, in its final form, to construct the engine on the compound principle, by constituting the left half of the cylinder the high-pressure portion, exhausting with the right-hand portion by suitable ports, thus completing the expansion.

The engine is self-enclosed and dust-proof, and cannot be tampered with. In the event of a breakdown it can be removed from its position and a spare engine fixed in less than fifteen minutes, the only attachments to the car or lorry frame being four bolts and caps, the removal of which enables the

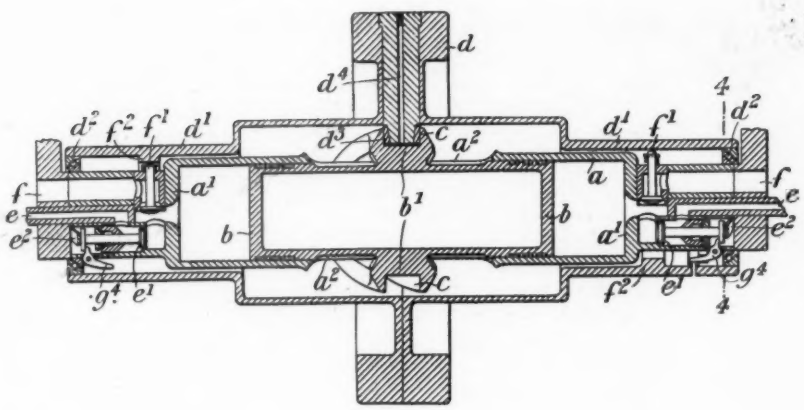


FIG. 1.

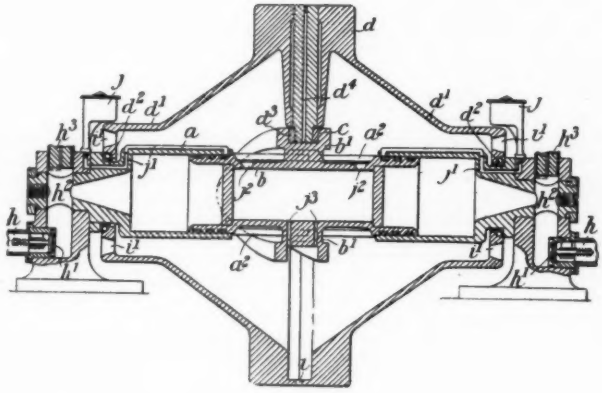


FIG. 4.

FIG. 2.

engine to be readily unshipped. The steam connections are not interfered with by removal, the bolting-down operation, it is claimed, sufficing to make all the necessary joints. Finally, it is stated that the cost of manufacture does not exceed £2 per h.p. for engines ranging in size from 5 to 15 b.h.p.

The inventor has adopted the same principle of converting reciprocating into rotatory motion in connection with an experimental internal-combustion engine. This, which is illustrated in Figs. 4 and 5, was not running at the time of our visit, but, we were told, had been at work, running up to 1000 revolutions per minute. In this engine an explosion takes place twice during each revolution of the fly-wheel. In construction it is identical in principle with the steam engine described above, and the foregoing description applies in so far as the cylinder, piston and fly-wheel are concerned; but in lieu of the inlet and exhaust passages and valves of the steam engine, each end of the cylinder is provided with an inlet passage *h* for the combustible mixture, which is forced in by pumps actuated by a cam movement not shown in the drawing. The pump is, however, seen in the photo, Fig. 4, as are the cam grooves, which are cut in the periphery of the cylinder casing. The flow of gas and air mixture is controlled by the valve h^1, which gives access to the combustion chamber h^2, wherein ignition takes place by the sparking plug h^3 in the usual

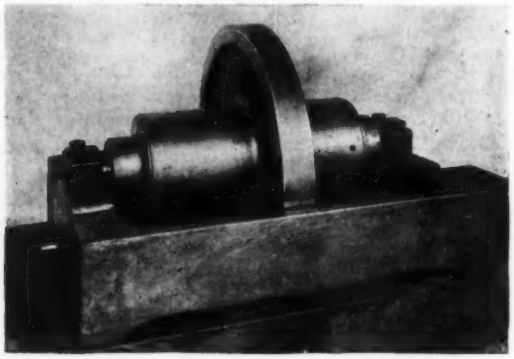


FIG. 3.—EXPERIMENTAL STEAM-LORRY ENGINE.

way. Exhaust takes place through the longitudinal slots a^2, a^2, and in the cylinder when the ends of the piston pass the ends of these slots, and exhaust gases passing into the annular space between the cylinder and the fly-wheel casing d^1. An induction is set up by the centrifugal action of the fly-wheel, which causes the products of combustion to escape through the outlets i, provided in the periphery. A considerable forced draught is also set up by the slots when the fly-wheel is revolving, which serves to cool the cylinder. Other modifications of this engine are proposed, viz., one operating on the Otto cycle and another in which an explosion takes place for each revolution of the fly-wheel. Again, in the case of the steam engine, should the efficiency warrant its construction on a commercial scale, several important advantages will be secured, notably by the simplicity of construction and the fact that all wearing parts are readily adjustable.

A New Blower.

MESSRS. JOHN GIBBS & SON, Duke Street, Liverpool, in their "Peerless Fan," have succeeded in producing a very efficient blower or exhauster fan for all purposes where vacuum or pressure, rather than volume, is required.

This type of centrifugal fan in many respects contradicts what has been considered to be the best and most efficient practice in this direction. The blades are constructed on a novel plan, with a result that the volume of air discharged per

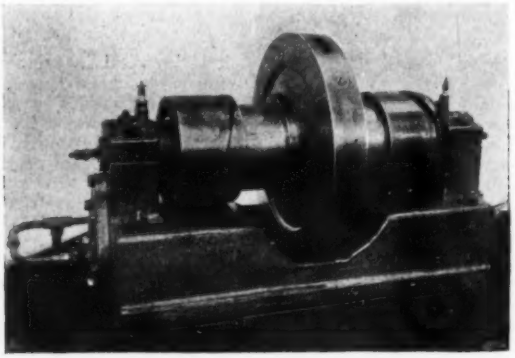
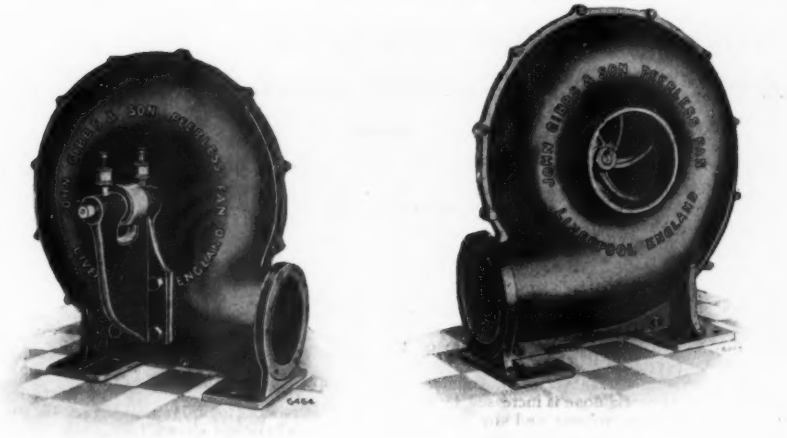


FIG. 5.—EXPERIMENTAL INTERNAL-COMBUSTION ENGINE.



GIBBS' "PEERLESS" FAN.

revolution, and the pressure produced when the area of discharge is restricted, is greater than many of the forms at present in vogue. Another feature of this machine is the absence of the detrimental back eddies which occur on the suction side of so many fans of this class, when they are working against pressure. Furthermore it is claimed that there is no end thrust on the fan-shaft even when the air is being drawn in upon one side only. This is a very remarkable feature, the advantages of which cannot be overlooked especially when the fans are direct coupled to electric motors. The bearings are of ample diameter for cool running at high speed, and owing to the small number of parts of which the fan is constructed—being in fact practically one piece — the bearings remain in perfect alignment both vertically and laterally. The performance of these fans, specially with regard to the small power consumed for the work done, renders them well worth the attention of prospective fan-users.

Abstracts from Important Articles and Papers.

STEAM.

The Thermal Effect and the Practical Utility of Superheated Steam. **Robert H. Smith.** In this paper, the author shows that the utility of superheating has no reason beyond that of keeping the engine cylinder walls and valve surface dry, the thermodynamic advantage being so very slight as not to compensate the expense of the process. Superheating always takes place at constant pressure—not at constant volume—and the mechanical work done is increased by expansion of the steam volume and by consequent enlargement of the $p\,v$ product per lb. of steam used. This work is done in the superheater and is transmitted forward to the cylinder piston.

If t_s be the absolute temperature of saturation at the given pressure, and t be the extra superheat, the ratio of expansion at constant pressure due to superheating is approximately

$$\frac{t_s + t}{t_s} = 1 + \frac{t}{t_s}.$$

Whatever be the index of the expansion curve after cut-off, the work done during admission and expansion is proportional to this $p\,v$. Taking ·48 as the specific heat of superheated steam, the amount of heat given to the steam per lb. to effect superheating is $\cdot 48t$. If H be the total heat of 1 lb. of saturated steam, the total heat of superheated steam is $H + \cdot 48t$, and superheat has increased this in the ratio $1 + \frac{\cdot 48t}{H}$. Comparing the ratio of increase of mechanical power with that of heat conducted, the ratio is

$$\frac{1 + \frac{t}{t_s}}{1 + \frac{\cdot 48t}{H}}.$$

In the following table of values of this ratio the superheat is 100 degs. Fahr. :—

$t_s =$	700	800	850 F° (abs.)
$p =$	24·5	117	218 lbs. per sq. in. abs.
$H =$	1,127	1,157	1,174 B.T.U. from 60° F.
$\frac{1 + \frac{t}{t_s}}{1 + \cdot 48\frac{t}{H}} =$	1·097	1·088	1·074.

The variation of this ratio is very slow, and for 10 degs. Fahr. superheat it is about 1 per cent. increase for low and ¾ per cent. increase for high pressures. Irrespective of back pressure, this would be the correct measure of the advantage gained by superheating if the superheat did not alter the index of expansion after cut-off, and if superheating did not cost anything beyond the additional supply of heat required for it. The above ratios agree very closely with the formula $(1{\cdot}1 - {\cdot}0001\,p)$. Apparently these small percentages of advantage would not of themselves justify the cost of superheating. But the index of the adiabatic curve of dry saturated steam is generally taken to be 1·13, while that of superheated steam is about 1·3; the latter giving much less work after cut-off; the work obtainable by saturated steam bearing to that obtainable by superheated steam the ratio $\frac{1{\cdot}13}{{\cdot}13} \times \frac{{\cdot}3}{1{\cdot}3} = 2$ (nearly). This reduces to insignificance the small 1 to ¾ per cent. advantage shown in the above table. But the fact is that, because of the physical character of the material used, adiabatic expansion, except when using superheated steam is a myth, the index 1·4 or 1·5 being more likely to be common, because, once condensation is started, it is impossible to prevent excessive condensation beyond what corresponds to adiabatic expansion. Again, recent experiments show much greater leakage past valve seats with wet than with dry steam. This is only to be expected, since leakage is a volumetric phenomenon, and if the fluid-solid frictional co-efficients which hinder it, were the same for dry steam as for hot water, then the same volume of hot water would leak through per second as of steam. For equal volumes the immensely greater weight of water than of steam is an ample explanation of the measured results of these experiments.—*The Electrical Review, Vol. LIV, No.* 1381, pp. 771–773.—*A. H. G.*

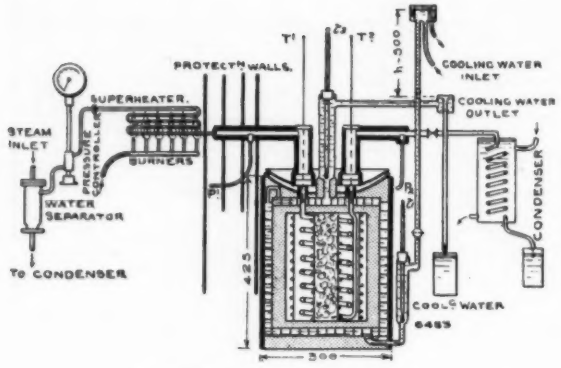
Specific Heat of Superheated Steam. **H. Lorenz.** The author has carried out extensive experiments with a view to determining the specific heat of superheated steam for invariable pressures within the limits of temperature and pressure important for industrial applications. A special process had to be developed, allowing of both the temperature and quantity of heat supplied to the steam being measured by means of electricity; any losses by conduction being obviated by having the calorimeter surrounded by a high vacuum and immersed in a paraffin bath of uniform temperature, which, at the same time, served to superheat the steam. Two years were required to complete the necessary electrical apparatus, in conjunction with Mr. E. Moeller of the Göttingen Institute for Technical Physics, the experiments being commenced in the autumn of 1903. As, however, special difficulties were experienced in fully carrying out the scheme contemplated, on account of the extreme sensitiveness of the whole arrangement, the author provisionally abandoned the electrical measurement of temperature. Steam,

after traversing a water-separator, enters the superheater through a pressure-reducing valve consisting of a copper tube, 12 mm. in clear width and bent several times, being heated by open Bunsen burners, the number of which may be readily controlled. In the immediate neighbourhood of this superheater, while being protected against its heat conduction and radiation by means of several sheets of asbestos pasteboard, there is placed the calorimeter, containing the steam coil covered with asbestos and surrounded on all sides by a water jacket. In the design of this calorimeter the experience gained in connection with the Junker fuel calorimeter could be utilised, a very regular water circulation being obtained by the use of overflow tubes. On both sides of the steam coil, there are electrical heating wires intended for supplying accurately measured amounts of energy to the steam, and thus reducing its temperature fall. The latter device, however, was not used in the present research. The thermometers introduced into the steam inlet and outlet tubes (long brass boxes being inserted in the intervals) reached as far as the entering and issuing points of the steam. In order to check the experiments, the two thermometers were exchanged, when no difference was noted. In connection with the water, special checking thermometers were used, divided in tenths of a degree. Before the main experiments were started, the heat exchange of the calorimeter with the surroundings was ascertained, when 2·4 heat units per hour (independently of the rate of flow of water) were found to be absorbed by the water for a temperature fall of 1 deg. between the water of the calorimeter and the surrounding air. Any necessary corrections of the thermometer readings were made, and the cover of the calorimeter was protected against any heat conduction or radiation from the connecting flange of the steam conduits by means of insulating material in addition to two sheets of asbestos pasteboard. The air layer between the latter was exchanged continuously by means of a strong air current. The original intention of extending the experiments to a number of pressures, starting from the saturated condition up to a superheat upwards of 400 degs., had to be partly abandoned on account of the impossibility of maintaining the joints made for high temperatures steam-tight; also for temperatures considerably below 200 degs., so that these temperatures had to be temporarily excluded. On the other hand, on exceeding 350 degs., the steam thermometers would begin oscillating so strongly as to make any state of equilibrium impossible. Readings were made every five minutes, while the cooling water and the steam, condensed in a tube condenser, were weighed every quarter of an hour. The figures given in the table are averages of experiments lasting about 30 mins. to 45 mins. each. The specific heat c_p required may be found from these figures by means of the following equation:

$$c_p = \frac{W(t_2 - t_1) - 2{\cdot}4\,(t_1 - t_w) + \Delta Q}{G(\tau_1 - \tau_2)}, \qquad (1)$$

where ΔQ is the water correction for the accumulation of heat in the calorimeter water (about 7 litres),

No. of Experiment	1	2	3	4	5	6	7	8	9	10	11	12
Steam temp. above τ_1 ... ... degs. Cent.	224·10	224·49	343·82	360·33	240·80	291·11	367·52	228·86	326·55	381·48	243·52	343·27
,, ,, below τ_2 ... ... ,,	190·99	190·71	244·77	281·71	203·96	232·90	260·08	181·96	262·40	296·30	161·59	244·70
Abs. steam pressure above p_1 ... kg./sq. cm.	7·009	6·970	6·970	7·095	2·200	9·125	9·170	4·486	4·398	4·186	2·071	2·107
,, ,, below p_2 ... ,,	6·649	6·616	6·670	6·795	1·735	8·730	8·870	4·136	3·778	3·686	1·793	1·671
Boiling temp. above δ_1 ... ... degs. Cent.	165·51	165·45	165·16	165·88	176·71	176·35	176·56	148·18	146·84	145·64	121·70	122·55
,, ,, below δ_2 ... ,,	163·15	163·08	163·40	164·15	174·50	174·41	175·19	145·21	141·94	141·07	117·10	115·90
Amount of steam in 1 hr. G ... ... kg.	8·270	8·214	5·992	6·276	7·152	5·821	4·136	5·648	7·676	7·068	3·206	4·633
Water temp. on entering t_1 ... degs. Cent.	4·970	5·215	5·258	5·565	4·944	5·162	5·780	5·109	5·220	5·043	5·466	5·561
,, ,, issuing t_2 ... ,,	13·250	13·580	16·476	18·422	14·798	16·445	19·004	13·247	19·26	25·146	18·490	18·164
Amount of water in 1 hr. W ... ... kg.	21·680	21·518	21·527	21·455	19·506	19·818	19·604	11·560	10·430	18·926	18·290	18·260
Water correction ΔQ ... ... heat units	+ 1·47	+ 0·88	+ 0·14	+ 0·60	− 2·10	+ 0·14	+ 2·28	− 0·84	+ 1·35	+ 7·63	− 0·63	+ 4·55
Mean water temp. t_w ... ... degs. Cent.	9·41	10·87	11·99	9·87	13·80	14·46	12·39	9·18	11·22	12·89	9·48	11·91
,, air temp. t_1 ... ... ,,	14·16	14·81	14·07	13·55	13·80	14·46	16·05	11·67	11·93	11·80	12·94	16·34
Air correction $2{\cdot}4\,(t_1 - t_w)$ ... heat units	− 12·12	− 12·98	− 7·68	− 3·74	− 9·43	− 8·78	− 8·74	− 5·97	− 2·71	+ 1·66	− 8·30	− 11·11
Absorb heat in 1 hr. ... ... ,,	161·86	167·90	233·94	272·72	180·68	214·97	252·74	152·37	235·08	284·45	137·83	221·84
Mean steam temp τ ... ... degs. Cent.	207·55	207·60	279·30	321·02	222·38	262·00	313·80	205·41	293·00	338·89	207·55	294·00
,, ,, pressure abs. ϕ ... kg./sq. cm.	6·829	6·816	6·820	6·945	8·968	8·908	8·808	4·311	4·053	3·936	1·932	1·889
Spec. heat according to equation (1) ...	0·617	0·605	0·564	0·552	0·685	0·634	0·568	0·575	0·477	0·473	0·484	0·524
Spec. heat according to equation (2) ...	0·640	0·625	0·572	0·558	0·715	0·640	0·571	0·591	0·491	0·483	0·538	0·500
Mean values of c	0·622		0·568	0·555	0·698	0·640	0·570	0·583	0·484	0·478	0·531	0·492



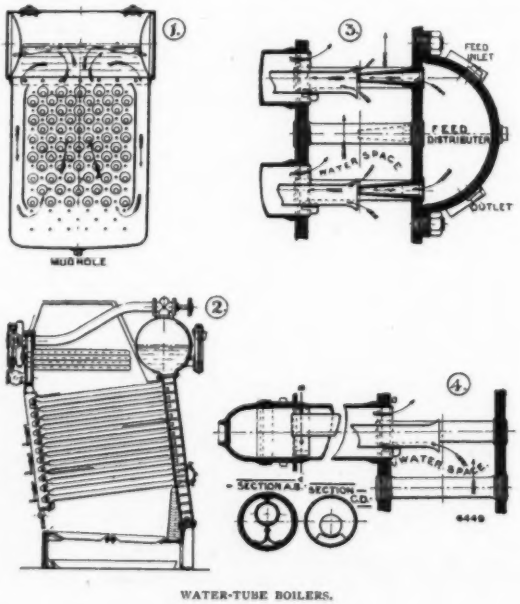
SPECIFIC HEAT OF SUPERHEATED STEAM.

while the remaining letters have the signification stated in the tables. The values thus calculated are recorded in the last line but two of the table. The same specific heats, as calculated in a second approximation from the equation:

$$c_p = \frac{W(t_2-t_1)-2{\cdot}4\,(t_1-t_w)+\Delta Q-0{\cdot}305\,G(\delta_1-\delta_2)}{G(\tau_1-\tau_2-\delta_1+\delta_2)} \qquad (2)$$

where the throttling effect is taken into account, are given in the last line but one. Comparing these results with the corresponding mean steam temperatures, it is seen that the specific heat of superheated steam doubtless increases along with the pressure, whereas starting from the range of saturation a decrease with temperature is noted, as stated by Lusanna (*Nuovo Cimento*, 1896) in the case of carbonic acid. For lower pressures the author's figures obviously approach the amount of 0·48, as found by Regnault for atmospheric pressures between 128 degs. and 217 degs.; while for higher pressures the figure of 0·60, as given by v. Bach, is satisfactorily confirmed.—*Zeitsch. Vers. Deutsch. Ing., Vol. XLVIII, No. 20*, pp. 698–700.—*A. G.*

Water-Tube Boilers. **M. E. Duchesne.** In the first part of this paper the author deals with a series of experiments which he carried out in order to ascertain the best inclination to the horizontal —ratio of length to diameter, etc.— of the tubes in a water-tube boiler. The tubes were of the Niclausse type, of glass, and the experiments were carried out at atmospheric pressure. With an inclination to the horizontal of 8 degs., a perfect circulation and a rapid disengagement of steam bubbles from the heated surfaces were obtained. The author has applied the results of these experiments to the design of a water-tube boiler (Figs. 1–4). The entrance of feed-water into the inner elements of the bottom rows of tubes as shown is found to materially aid circulation. Fig. 4 shows the method of construction adopted for the tubes of this boiler, this construction having proved very successful in actual working. The diameter of the water-tube is 3·1 ins. with a maximum length of thirty times this diameter, experiments showing that by using a greater ratio of length to diameter than this, the circulation is unduly retarded. — *Memoir Soc. Ing. Civils de France, Series 6, No. 1*, pp. 59–83.—*A. H. G.*



WATER-TUBE BOILERS.

METALLURGY.

The Detection of the Finishing Temperature of Steel Rails by the Thermo-magnetic Selector. **A. Sauveur** *and* **J. Whiting.** Steel, on cooling from a high temperature, remains non-

magnetic until a certain critical temperature is reached, which, in the case of medium and high-carbon steels, occurs at the point of recalescence; this point also marking the end of the crystallising range. The lower the finishing-temperature of steel the finer the structure; hence rails, in order to possess a fine structure, should be magnetic when leaving the finishing-rolls, or become so shortly after leaving them, thus indicating that they were finished but slightly above the point of recalescence or crystallisation. In order to detect the finishing-temperature the authors have devised a magnetic test by means of which magnetic rails are marked, non-magnetic ones receiving no stamp. The device consists of a rail-stamping machine placed in the runway from the hot saws. The rails pass over two rollers keyed on to steel shafts extended back to the frame of the machine (Fig. 1). The action of this machine is as follows: When a rail which is cold enough to be magnetic passes over the two rollers it closes the air-gap between them, and by closing this air-gap the magnetic flux passing through the steel shafts, due to the current flowing in the solenoid, is increased many times. By connecting the two steel shafts by means of a magnetic rail, a horse-shoe magnet is formed with the poles at the rear of the machine. This increase in flux causes the keeper bridge-piece to be attracted, thus forming a complete magnetic circuit through the two steel shafts, the rail and the keeper. As soon as the keeper is attracted it closes the electric switch, which establishes a circuit through the electric motor, which in turn revolves the shaft carrying the stamp, thereby causing the stamping of the magnetic rail. Rails which are finished too hot to be magnetic pass through without being stamped.—*Proc. Am. Soc. for Testing Materials and Iron and Steel Mag., Vol. VII, No. 5*, pp. 503–507.

Electrolytic Iron. **C. F. Burgess** and **Carl Hambuechen.** After many preliminary experiments ferrous sulphate, with a certain percentage of ammonium sulphate, was adopted as yielding the most satisfactory results, and the process, at present in operation on a small experimental scale, is briefly described as follows: The electrolyte consists of ferrous and ammonium sulphates; the current density at the cathode is 6amps. to 10amps. per sq. ft. of cathode surface, and at the anode slightly less. The electro-motive force for each cell is slightly under 1 volt, and the temperature of the electrolyte is about 30 degs. Cent. The anodes are of wrought-iron; the starting sheets for the cathode are of thin sheet-iron previously cleaned of rust and scale. Serious difficulties were encountered in obtaining thick deposits of iron, but improvements have been gradually effected until at the present time it is possible to continue the run four weeks without replacing the cathodes. At the end of this period the cathode reaches an average thickness of $\frac{3}{4}$ in., the surface then being so rough and nodular that it is not desirable to continue the deposition further. The current efficiency is nearly 100 per cent., or a deposition of about 1 gram per amp.-hr. The electro-motive force being 1 volt, a deposition of about 22 lbs. of iron per kw.-hr. is obtained. For the past six months three tanks have been kept in almost continuous operation; these tanks are 8 ft. wide by 13 ft. long by 15 ft. deep, and contain two anodes and one cathode. The cathodes from the tanks weigh about 20 lbs., and probably form the largest pieces of electrolytic iron yet produced. The total amount of material produced is about half a ton. Under these experimental conditions the cost of refining, aside from fixed charges on plant and with power at 30·00 per kw.-year, would appear to be under $\frac{1}{2}$ cent. per lb. of iron. Should these figures be realised in practice the value of the product will still depend on its utility—or, in other words, the properties of the refined metal. Analyses of the iron obtained have shown its purity to exceed 99·9 per cent. No carbon could be detected, and silicon and manganese appear to be absent. Hydrogen is, however, present in appreciable quantity, and it appears to have a decisive influence on the metal The iron as deposited is hard, but on heating the hydrogen is given off and the metal becomes soft; whilst, after raising to a welding-heat, the malleability and toughness are similar to those of Swedish iron. Experiments have shown that the hysteresis, permeability, and electric resistance of electrolytic iron are influenced by the amount of hydrogen it contains.—*Paper read before the Am. Electro-Chemical Soc., Iron Trade Rev., Vol. XXXVI, No. 19*, pp. 40–42.

The Effects of Strain and Annealing. **Wm. Campbell, Ph.D.** The structures presented by the surfaces of small ingots and buttons of aluminium, antimony, bismuth, cadmium, copper, gold, lead, platinum, silver, tin, and zinc have been studied. The effects of slight strain have been observed by bending the pieces to a slight angle and straightening again; whilst the influence of heavy strain on the structure has been studied after hammering or rolling. Annealing was conducted by heating the specimens in an air-bath or on a hot plate at a temperature of 200 degs. Cent. From his studies the author concludes that the effect of strain is to produce a parallel slipping along definite directions, usually related to the orientation of the primary crystals and also to the direction of strain. In certain cases the effect is seen in the formation of one or more systems of parallel slip-lines; in others in the production of parallel systems of twinning. As Ewing and Rosenhain have pointed out, the first slip-lines are perpendicular to the direction of straining; whereas further straining produces other systems. When the strain has been severe—as, for instance, by rolling—the large primary crystals are broken down and a finer crystallisation takes their place; the greater the mechanical work upon the metal the finer the resulting crystallisation. Annealing causes the fine crystallisation to rearrange itself, and produces the growth of crystals, whose size apparently depends to some extent both on the time and temperature of annealing and the thickness of the metal. In the case of lead and tin this arrangement was observed to take place at ordinary temperatures, but at a comparatively slow rate.—*Chicago Meeting of the Am. Soc. of Mech. Engrs*, pp. 1–38.

Tungsten Steels. **Léon Guillet.** Two series of steels of ascending content of tungsten have been examined, the first series being comparatively low in carbon, whilst the second one contains approximately 0·8 per cent. The actual amount of carbon and tungsten present are given in the accompanying table, together with the results obtained from tensile, shock, and hardness tests :—

Series I.

Carbon %.	Tungsten %.	Elastic Limit.	Maximum Stress.	Elongation %.	Frémont Test. Kilogrammètres.	Hardness. Brinell's.
0·117	0·412	30·1	41·1	19	25	97
0·113	0·939	31·6	42·1	18	29	97
0·110	1·750	37·6	48·9	18	26	109
0·126	4·965	33·9	63·2	13·5	20	153
0·176	11·890	79·1	86·6	5	6	223
0·201	14·270	54·6	77·2	10·5	5	167
0·221	20·710	48·9	71·5	9	4	163
0·219	24·350	41·4	60·2	4	6	187
0·276	27·750	56·4	67·0	2·5	6	217

Series II.

Carbon %.	Tungsten %.	Elastic Limit.	Maximum Stress.	Elongation %.	Frémont Test. Kilogrammètres.	Hardness. Brinell's.
0·861	0·397	56·4	103·2	6·0	1	241
0·852	0·951	62·8	113·0	5·5	1	241
0·795	2·750	75·3	124·3	5·5	1	302
0·823	4·676	85·8	126·5	5	5	302
0·815	9·991	101·7	134·8	3·5	4	293
0·715	14·739	67·8	129·9	2	5	351
0·797	19·250	71·5	105·4	3·5	6	277
0·743	25·270	94·1	113·0	0·5	6	351

—*Revue de Métallurgie*, pp. 263–283.

Chemical Classification and its Application to the Constituents of Steel. **Professor Le Chatelier.** The study of alloys presents no little difficulty in that individual crystals of different species of homogeneous matters, definite compounds or solutions, are often exceedingly minute, and at times even so small as to escape the resolving powers of the most powerful microscope. On the other hand, the study of natural rocks is more favourably placed, in that the constituent crystals are larger and occasionally visible to unaided vision. With metallic alloys, particularly those of iron, similar conditions do not exist, and a classification of constituents which do not permit of isolation has to be made. The difficulties thus presented are intensified by the multiplication of methods; thus, in combining the microscopical examination after attack with various etching mediums, with measurements of electrical resistance, dilatation, absorption of latent heat, etc. Commencing with pure iron, which microscopically presents a series of grains of no definite geometrical contour, Osmond has shown that it may exist in different allotropic states. For instance, at 900 degs. Cent. he proves a transformation characterised by the absorption of latent heat, by a sharp contraction, by changes in the laws of thermo-electric force, and electric conductivity, and by a sharp, but less marked, change in the magnetic properties. A further transformation is found at 750 degs. Cent., but of a nature quite distinct from that occurring at 900 degs. After iron the second essential constituent of steel is carbon, which is known under three allotropic states of diamond, graphite, and amorphous carbon. In the alloys of iron the variety graphite only is met with. A definite carbide of iron Fe_3C or "cementite" is formed by the association of iron with carbon, and this compound forms the major part of white cast-iron. Iron possesses the property of forming solid solutions with many substances, occasionally in all proportions, as with manganese and nickel, and with others to a limited extent only, as with phosphorus up to 1 per cent. and silicon up to 15 per cent. Iron containing in solid solution slight traces of these substances constitutes the phase "pure iron" in ordinary iron and steels which are always slightly impure. This phase is termed "ferrite." In a similar manner cementite can mix isomorphically with the carbide of manganese Mn_3C. These features illustrate in the case of iron the difficulties already noted. Carbon yields with iron solid solutions of great metal-

lurgical importance, but of very difficult study. These solid solutions are only normally stable at temperatures superior to 700 degs. Cent., and they may only be preserved in an incomplete manner at ordinary temperatures by rapid cooling. Solid solutions of iron and carbon possess when fluid all the properties of liquid solutions, and the carbon diffuses like salts in their aqueous solutions. These solutions present the phenomenon of saturation—they deposit by cooling, as the case may be, ferrite, cementite, or even graphite. The curves of solubility determined by Osmond & Roberts-Austen show that the addition to pure iron of an increasing content of carbon gradually lowers the transformation point until with 0·9 per cent. carbon it is lowered from the normal one of 900 degs. Cent. to 700 degs. This temperature of 700 degs. Cent., known as the recalescence, corresponds to the eutectic point of the alloys of minimum fusion. The solidification of this eutectic gives simultaneously alternate plates of iron and cementite, termed "pearlite." The classification of the constituents of quenched steels cannot be considered definite. The first supposition leads to the conclusion that the effect of quenching is to preserve at ordinary temperatures a normal stable variety of the high temperatures. With hardened steels, however, this is not the case; for, though their condition is entirely distinct from the stable cold state, it is, nevertheless, a condition not absolutely identical with that existing at the high temperatures. For instance, steel at high temperatures is no more magnetic than the variety of pure iron stable above 900 degs., yet quenched steels are very magnetic and are employed in the construction of magnets. Further, a single variety is not obtained by quenching, but several constituents may be obtained by modifying the quenching temperature or varying the content of carbon. The study of these quenched steels is difficult, owing to the impossibility of isolating the constituents in order to examine their individual properties. Thus one section may contain several constituents, and the only method by which they may be studied is the microscopical examination of polished and etched surfaces. The most familiar etching mediums employed in differentiating the constituents of quenched steels are solutions in absolute alcohol of azotic, hydrochloric, or picric acids. Osmond has distinguished in these steels three constituents, termed respectively "troostite," "martensite," and "austenite." In steels rapidly cooled, but not actually quenched, or in quenched steels lightly annealed, a fourth constituent ("sorbite") is found. Troostite is most familiar as a dark border separating pearlite and martensite, and may be obtained by quenching steels in the middle of their transformation or about 750 degs. Cent. Martensite is obtained by quenching steels of variable composition (0·3 to 1·3 per cent. carbon) from above the recalescence point. A steel of 1·4 per cent. carbon quenched from 1,200 degs. Cent. and etched with picric acid in alcoholic solution presents austenite. The differentiation of these three constituents rests solely on differences of feeble attack by the etching medium employed.—*Revue de Métallurgie, Vol. I, No. 4*, pp. 207–225.

Experiments with a Lathe-tool Dynamometer. **J. C. Nicholson, D.Sc.** In the tool-steel trials made by the Manchester Committee in 1902–3,* there appeared an entire lack of uniformity in the shapes and angles of the tools submitted for test. There was also no obvious connection between the shapes and angles of the tools and the cutting-forces upon these tools deduced in the report from the electrical-power measurements made by the Committee. Neither did the shape or angle supply a clue to the causes of success and failure in the various trials with different tools. The necessary reconsideration of the design of lathes for the rapid and heavy cutting rendered possible by the new tool-steels, calls for a thorough and systematic investigation of the forces acting upon a cutting-tool, and the author has undertaken a somewhat extensive research into the matter, involving over 300 serial trials. The opinion generally held concerning the relation of the cutting force to the depth of cut was that—as first enunciated by Hartig—the former varied in simple proportion to the latter. Although this law of variation was controverted by Prof. R. H. Smith in his work on "Cutting Tools," the author of this paper finds that his experiments substantiate Hartig's law to a first approximate. The apparatus used consisted of two specially arranged dynamometers capable of measuring forces up to 15 tons on the tool-point when taking a cut; the force-measurer itself consisting of an hydraulic support and a Bourdon gauge. The results of the trials are given in very complete tabulated form, from which a series of interesting curves have been plotted. The chief conclusions deduced from the experiments are as follows: In the case of medium cast-iron the variation of the cutting-stress with the cutting-angle is very marked. It varies by nearly 100 per cent. of its smallest value, which takes place, in every case, for a cutting-angle of about 60 degs. This angle of minimum cutting-force is by no means that of greatest durability. A cutting-angle of 80 degs. is that indicated as being best for shop use, and the cutting-stress for this angle is about 75 tons per sq. in. In the case of soft steel the variation of the cutting-stress with the traverse in the case of soft steel is somewhat complicated. For keen cutting-angles (below 75 degs.) fine traverses require less cutting-force than wide ones; whilst for blunt-nosed tools (*i.e.*, cutting-angles greater than 75 degs.) the reverse is the case, and the fine traverse-cut requires the greater effort to remove. At a cutting-angle of 75 degs. the stress is the same whether the traverse be $\frac{1}{16}$ in. or $\frac{1}{8}$ in., and has the value of about 100 tons per sq. in. It is curious to remark that this angle of 75 degs. is also about the best angle for shop use. Ordinary shop-tool, when cutting soft steel of this quality, exerts a vertical force of 100 tons per sq. in. of area of cut removed, irrespective of the proportion of width of traverse to depth of cut. [It may be pointed out that 98·5 tons per sq. in. was obtained in the Manchester experiments as the average cutting stress for the endurance trials on soft steel

* Cf. THE ENGINEERING REVIEW, December, 1903, to April, 1904.

(Table 20 of Report), in which one shape of tool was used throughout, the cutting angle being about 70 degs. The stress on these endurance trials ought to have been about 88 tons, or even less, as the speed was 90 ft. instead of 50 ft. per min.; but it must be remembered that the electrical method of measuring the cutting force from which the figure 98·5 was deduced includes not only the vertical work, but also that done in pushing away the shaving over the face of the tool, and ought in most cases to give a greater value for the cutting stress than that attained with the dynamometer. The agreement is therefore very close, and the two results are mutually confirmatory.] The experiments made for examining the durability of different cutting angles showed that a cutting angle of from 75 degs. to 80 degs. with tools of 45 degs. plan angle were the most durable for medium cast-iron. As the cut ($\frac{1}{16}$ in.) was somewhat shallow, and the tool-point had a small radius (about $\frac{3}{32}$ in.) in plan, the shaving moved off in a direction nearly perpendicular to the axis of the work, instead of at right angles to the cutting edge of the tool (45 degs. in plan). This means that the actual cutting angle, measured in the direction of motion of the shaving (in plan) [or the true cutting angle as per Manchester report, p. 246] was about 81 degs. Tools should therefore be ground for maximum endurance in the cutting of cast-iron in ordinary shop practice, so that their true cutting angles are about 81 degs.; or, if they are allowed 6 degs. clearance for working on the level of lathe centres, they should have an included angle of about 75 degs. In the case of medium fluid-pressed steel the trials seemed to show that a cutting angle of about 70 degs. (included angle 65 degs.) is that which will last the longest in rapid cutting. The plan angle of the cutting edge was 45 degs. throughout.—*Proc. Am. Soc. Mech. Engrs., May-June.—A.*

Gases in Pig-iron. **E. Münker.** The author has collected and analysed the gases evolved from a series of molten pig-irons. In all, some 67 analyses are reported, and of these the following are selected:—

	No.	H.	CO.	CO_2.	O.	Remarks.
FORGE PIG-IRON	1	37·1	25·2	1·2	0·4	Fracture white.
Analyses:—						
C. 2·0 to 3·5 P. 0·2 to 0·3 ...	7	9·9	11·2	7·4	0·4	
Si. 0·4 to 2·0 S. 0·01 to 0·06 ...	14	43·5	17·0	2·6	0·2	
Mn. 2·0 to 6·5 Cu. 0·1 to 0·3 ...	23	18·0	18·4	7·2	0·4	
STEEL-MAKING IRON	24	12·8	30·8	0·6	0·6	Fracture radiant and glassy.
Analyses:—						
C. 3·0 to 3·5 Si. 0·4 to 0·7 Mn. 4·0 to 6·0	28	47·5	32·8	0·8	0·6	
P. 0·05 to 0·08 S. trace to 0·02 Cu. 0·1 to 0·3	33	14·4	33·8	0·4	0·0	
SPIEGELEISEN	34	19·2	14·6	0·2	0·0	11·4% Mn. Fractures glassy.
Analyses:—						
C. 3·5 to 4·5 P. 0·05 to 0·1 ...	40	10·2	16·8	0·2	0·4	14·3 ,,
Si. 0·3 to 1·0 S. trace to 0·01 ...	46	52·1	10·2	0·0	0·2	22·8 ,,
Mn. 11·0 to 23·0 Cu. 0·2 to 0·3 ...	48	34·8	12·0	0·2	0·2	23·1 ,,
Analyses:—						
C. 3·5 to 4 P. 0·12 to 0·16 ...	49	73·7	12·6	0·6	0·4	Fracture grey, with fine grain.
Si. 1·5 to 2·5 S. trace to 0·02 ...	50	82·9	4·0	1·0	0·4	
Mn. 2·0 to 4·0 Cu. 0·15 to 0·25 ...	53	18·2	15·0	1·6	0·8	
FOUNDRY IRON	55	8·3	4·8	7·8	0·8	Fracture grey, and fine grain.
Analyses:—						
C. 3·8 to 4·2 P. 0·4 to 0·7 ...	60	10·0	4·0	8·2	0·4	
Si. 2·0 to 3·5 S. trace to 0·04 ..	64	2·5	1·8	8·2	0·6	
Mn. 0·4 to 0·8	67	7·7	6·8	12·2	0·6	

As foundry iron contains the most silicon, it would almost appear that the extraction of hydrogen is prevented by the silicon present; but, on the other hand, the forge quality also contains appreciable

contents of silicon, and yet evolves considerable quantities of hydrogen. Manganese tends to promote the occlusion of hydrogen, and, if manganese were alone responsible for this occlusion, the Spiegeleisen should develop the most hydrogen; but, as results show, this is not the case. A comparison of the figures given, therefore, leads to the supposition that the chemical composition of the iron does not determine the character of the gases evolved on solidification. A greater part is doubtless played by the temperature of the metal in the hearth, the degree of silica formation, the fusibility of the slag, and the amount of moisture introduced into the furnace by the blast. The amount and character of the gases are also, to some extent, a factor of the fluidity of the molten iron. Turning to the gases collected from the solidified metal, the following table shows some results obtained:—

GASES IN SOLID PIG IRON.

	No.	Capacity of drilled hole. c. cm.	Gas at normal pressure.		H.	Co.
			C. cm.	In percentage of drilled hole.		
Foundry iron ...	1	41	13·4	33	47·7	Not fixed.
,, ,, ...	4	49	30·0	61	68·5	,, ,,
,, ,, ...	8	58	47·6	82	66·1	0·6
,, ,, ...	10	48	17·0	35	59·0	3·5
,, ,, ...	16	62	56·8	92	57·3	1·05
,, ,, ...	18	53	18·8	35	53·3	4·4

The differences in the volumes of the gases collected from similar pieces of metal show that an average value cannot be attached to the foregoing figures. This variation in volume is due to the irregular distribution of blowholes.—*Stahl und Eisen, Vol. XXIV, No. 1*, pp. 23-27.—*P. L.*

TRACTION.

The Resistance to Motion of Railway Vehicles and the Performance of Locomotives. **Von Borries.** In connection with the electric high-speed railway experiments recently made in Germany, the question as to how far steam locomotives would allow of higher speeds being reached has been frequently discussed, and the author proposes to contribute to the solution of this problem by investigating the resistances to motion, from which the possibilities of such railways may be ascertained. Any formulæ so far used in this connection failed to give available values for speeds exceeding, say, 100 km. Now the motion-resistance of a moving train may be ascertained by determining, when the train is running at a uniform speed, the efficiency of the locomotive by means of an indicator diagram, thus allowing the "steam pull" (Dampfzugkraft) required for the whole train to be calculated, the pull of the train itself being found by means of a dynamometer fixed on the tender; the difference of both pulls will represent the total resistance of the locomotive. This process, however, offers certain drawbacks on account of the difficulty found in running at a uniform speed. A second method consists in allowing the trains to run to a standstill after the steam has been shut off, calculating the acting force from the variation of the speed, when the running resistance will be equivalent to the retarding force as calculated from the decrease in speed plus the gradient-thrust. This process will give values the more accurate as the running-out speed is more uniform. An objection to this method is, however, the fact that the pistons of the locomotive, in running at no load, act like pumps, and thus set up considerable resistance. In order to eliminate piston resistance the author commenced in the summer of 1901 some experiments in Hanover with 2-4-coupled compound high-speed locomotives of the Prussian State Railway, these experiments being made: (1) with the engine in ordinary working order; (2) with the valve-boxes taken off, and the cylinders heated by steam; and (3) with the motion disconnected. The locomotive experimented upon was pushed by another in the way mentioned under (2) or (3), when, after the desired speed was reached, steam was shut off in the rear locomotive, and both would begin running down a gradient at no load, the former being in advance of the other. The running resistance with the driving-mechanism disconnected was soon found to be much lower than hitherto supposed, being not much above that of a coach. In fact, this resistance, for low speeds, was only about 1·6 kg. per one ton; while with speeds of 50 km. and 100 km. per hour respectively, figures ranging betweeen 3 kg. to 3·5 kg. and 6 kg. to 7 kg. per ton were found. With locomotives in full working-order these resistances were doubled; and, with only pistons running, figures about 1·5 the above value were obtained. Similar experiments were made, on the author's suggestions, with the Erfurt Railway Department by Mr. Leitzmann, when the following formulæ were found to represent the average results:—

(1) In working order

$$w_1 = 2{,}7 + 0{,}045\ V + 0{,}0004\ V^2$$

(2) Without driving mechanism

$$w_1 = 1{,}5 + 0{,}027\ V + 0{,}0004\ V^2$$

The resistance due to the driving mechanism was therefore $1{\cdot}2 + 0{\cdot}018\ V$. The formula (2) may be used for calculating the pull at the periphery of the driving wheel, when a certain percentage has to be allowed for friction in the engine; but this method is hardly practical. When allowing for the friction of the working driving-mechanism in working order (according to Prof. Goss' measurement made on a locomotive of Purdue University), the following formula is found for the total resistance of the working locomotive:

$$w_1 = 4{,}3 + 0{,}027\ V + 0{,}0004\ V^2$$

According to observations made in connection with the high-speed electric cars, the air resistance is $0{\cdot}0052\ V^2$ for 1 square metre of rectangular front surface, being somewhat higher for the maximum speed of 200 km. Comparing this value with Barbier's results, and with recent observations by Nadal ("Revue générale des Chemins de Fer, 1904, p. 196"), the author thus modifies the above formula:

$$w_1 = 4 + 0{\cdot}027\ V + 0{,}0007\ V^2$$

Instead of 4 kg. somewhat higher values, corresponding with a piston pressure of about 0·25 atmosphere, would have to be adopted for locomotives having larger pistons. As, however, the above formula for modern types of locomotives upwards of 80 tons in weight, will give too high values with V^2, the front surface increasing only to a small extent along with the weight, the front surface of the first wagon projecting beyond the tender should be added to that of the locomotive. The motion resistance of the cars was found by Mr. Leitzmann for a train of 40 axles, comprising two axle-cars, each axle being loaded with about 6 tons, to be

$$w_2 = 1{,}3 + \frac{V}{247} + \frac{V^2}{1470}$$

According to the high-speed trials, the running resistance, being lowest for $V = 5$ km. (1·44 kg.), would increase for $V = 40$ km. up to 2·05 kg., the increase being slower for augmenting speeds (for $V = 120$ km. the resistance is 2·9 kg. per ton); taking into account these results as well as the experiments of Nadal, Frank and others, the following formula is proposed:

$$w_2 = 1{,}5 + 0{\cdot}012\ V + \left(\frac{0{,}3}{q} + 0{,}2\right)\frac{V^2}{1000}$$

From these formulæ the following figures are found for trains consisting of four-axle cars 30 tons in weight:

V = km.	0	20	40	60	80	100	120
Leitzmann	1·2	1·5	2·2	3·2	4·6	6·5	8·5
Frank	2·5	2·6	3·0	3·6	4·4	5·5	6·8
Barbier (deducting 1/3)	1·4	1·6	2·0	3·0	4·3	5·7	7·6
Von Borries ...	1·5	1·9	2·5	3·3	4·3	5·7	7·2

As regards the performance of locomotives for high speeds, Mr. Frank uses a formula according to which each sq. m. of heating surface in contact with the fire will give at the periphery of the driving wheel an output of $0{\cdot}617\ \sqrt{V}$ h.p. This formula does not seem to be available for speeds of over 90 km., the values then obtained being too high. A high-speed locomotive recently built by Messrs. Henschel & Son, Cassel, gave, for instance, an output of 6·6 h.p. per 1 sq. m. heating surface, this locomotive readily conveying a train of 300 tons at a speed of 120 km. A 2-6 coupled locomotive of a heating surface of 260 sq. m., such as the Henschel engine, seems in fact to afford the limit of suitability. This locomotive will allow of an average speed as high as 110 km. being obtained on favourable tracks. Making the front surfaces of the locomotive wedge-shaped affords a most simple and inexpensive means of improving the performance of locomotives, angles as sharp as possible being chosen.—*Zeitsch. Vers. Deutsch. Ing. Vol. CCCXLVIII., No. 22*, pp. 810–13).—*A. G.*

Aërial Navigation. **J. Massau.** In this paper the author discusses mathematically some of the principles of aërial navigation. He classifies flying machines in three divisions, according as the sustaining and elevating force is produced—(1) by alternating motion, as of the wings of birds; or (2) by propellers revolving round vertical axes, as in helicoptera; or (3) by surfaces inclined to the horizon and animated by a horizontal translational movement, as in aëroplanes. From the fundamental equations of elasticity he deduces the two following "principles of similitude": (1) Given a system in equilibrium, we can construct another geometrically similar, in which the elastic forces vary as the dimensions and the absolute forces as the cube of the dimensions; and (2) we can also construct another system geometrically similar in which the dimensions are multiplied by a, the elastic forces by b, and the absolute forces by ba^2. The author then discusses the application of these principles to flying machines, and analyses mathematically the relations between their dimensions and the weight of the motor, the resistance to motion and the stability of the ship, the size of propellers and the maximum velocity of the ship, and the efficiency.—*Ann. Ing. Gand, 1903, Vol. II*, pp. 370–382.—*W. C. H.*

MISCELLANEOUS.

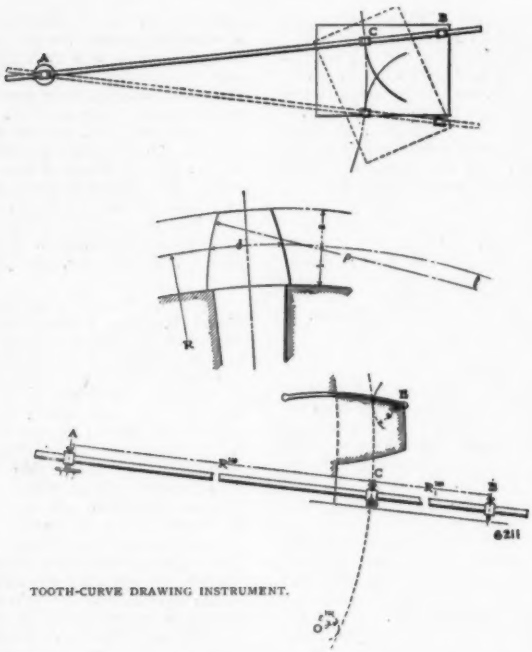
Asphalt Reservoir Lining. **John W. Hill.** In the course of a paper which deals with the construction of the Belmont Filtration Works for the improvement, extension, and filtration of the water supply of West Philadelphia, the author describes the method of ensuring water-tightness of the asphalt lining to the sedimentation reservoir, the area of which is nearly 11 acres. Over the concrete floor and on the slopes to a height 10 ft. vertical below the water-line, a lining of asphalt $\frac{3}{4}$ in. thick was placed in two layers, each uniformly $\frac{3}{8}$ in thick. This lining is in

addition to the outer lining of 18 ins. of puddle and 6 ins. of concrete floor and slope paving, as it was thought necessary to make the reservoir as nearly water-tight as such structures can be made. Two mixtures of asphalt were used, containing as an average the following weight of materials for each batch that went into the kettle:—Asphalt mixture used on the floor of reservoirs and in the first layer on slope: 585 lbs. Seyssel mastic, 315 lbs. grit, 50 lbs. refined Trinidad asphalt, 50 lbs. refined Bermudez asphalt. Asphalt mixture used in second layer on slopes of reservoir: 598 lbs. Seyssell mastic, 332 lbs. grit, 33 lbs. refined Trinidad asphalt, 37 lbs. Bermudez asphalt. The above mixtures gave an average of 15·5 per cent. of bitumen for the lining on the floor and first layer on the slopes, and 13·2 per cent. of bitumen for the second layer on the slopes. On the floor of the basins the asphalt was laid on the smooth concrete, but on the slopes the concrete was roughened by indenting grooves ½ in. deep and ¾ in. wide, spaced about 4 in. centre from the toe to the top of the asphalt line, to secure the asphalt against slipping or creeping on the concrete.—*Jnl. Franklin Inst., Vol. CLVII, No. 1*, pp. 1–31; *Vol. CLVII, No. 2*, pp. 89–110; *and Vol. CLVII, No. 3*, pp. 179–199.

Devices for Removing Chips and Dust. **Dr. Prandtl.** The author divides the different forms of apparatus used for removing chips and dust in several groups according to the class of workshops they are intended for. These devices afford the advantage of providing better air in the workshops and a longer life to the machines, while, in the case of wood-working factories, there is the further advantage of a better utilisation of the room. The amount of energy necessary for dealing with heavy shavings and sawdust is relatively high as compared with that necessary in the case of dusty air. In the wood-working machine shops of the Maschinenbau-Gesellschaft, Nürnberg, 160 h p. was formerly used for operating the machine tools, and a further 100 h.p. for operating the suction devices. After the power-saving devices invented by the author had been installed, the consumption of energy was reduced down to 35 to 40 h.p., the amount of air sucked off being about 25 cub. m. per sec., and the amount of chips dealt with daily as high as about 10,000 kg. In these devices the amount of air sucked off is the one just necessary, and the chips are drawn off in their natural flying direction. As the suction energy decreases as fourth power of the distance from the ends of the conduit, the device should be placed at a distance from tools as small as possible, the suction intensity being adapted to the kind of chips in question, and their possibility of transportation. For the movement of dusty air, wide tubes and small speeds of the air appear to be economical, the tubes being placed in a steep position, while, for transporting shavings, high speeds ranging between 7 and 20 m. per sec. are necessary. In order to avoid any unnecessary waste of energy, the tubes should be smooth and as little curved as possible, several tubes being connected at an angle as sharp as possible. In the case of dusty air, any description of blowing apparatus is equally suitable, while centrifugal blowers with sheet-iron plates are necessary for dealing with chips. As regards the depositing devices, there are the dust chambers where, the air being arrested, the dust is made to fall down, the filters, and the centrifugal cleaning devices. In the cleaning device designed by the author, the drawback of the high back-pressure produced by the rotating air is obviated by fitting in a kind of turbine wheel.—*Paper read before the Hanover Section of the Verein Deutscher Ingenieure;* see *Zeitsch. Vers. Deutsch. Ing., Vol. XLVIII, No. 13*, pp. 458–459.—*A. G.*

Notes on the Packing of Machinery for Export. **Paul Roux.** Packing comprises the series of operations which have for their object the preservation of the machine in good condition from the time it leaves the producing factory to the time of final delivery at the works of the purchaser, and the care with which these operations are carried out, affects largely not only their cost, but expenses of transportation, etc., which may amount to a considerable percentage of the selling price. *Preparation of Machines.*—After careful inspection all machine tools should be more or less dismounted, and perhaps the most important consideration here is its volume or cubage when packed for shipment. Marine freights are generally reckoned per 40 cub. ft., so the exporter must see that the weight does not occupy more than 40 cub. ft. per ton. *Dismounting Fragile Parts.*—All fragile projecting parts should be removed and tagged with labels, fully explaining position on machine. All delicate parts should be protected against rough handling. *Preservation.*—Finished parts should be coated with some preparation which should preserve against rust, but not attack the metal. *Packing Cases.*—These must fulfil two requirements. Effectually protect the machine against all shocks and injury, and facilitate the handling of the machine, and provision must also be made for the examination of the machine in the Custom House. *Lists and Drawings* should accompany the machine, carefully enveloped in waterproof paper, and tacked inside the case. *Exterior Marks* for identification should be very clear, and the name of port of discharge should be in letters at least 2 in. high. Finally, exporters are urged to paint a black circle round the heads of all nails and screws, which should be removed, in order to unpack the machine with the least work and without injury to any parts.—*Paper contributed to the American Chamber of Commerce, Eng., Vol. LXXVII, 1891*, pp. 313-314.

Tooth-Curve Drawing Instrument. **F. Hass.** Fig. 1 illustrates the author's tooth-curve drawing instrument. To a beam compass centered at *A* (Fig. 1), we pivot at *B* the piece of paper on which the tooth outline is to be traced. At *C*, the point of contact of the pitch circles, the beam compass carries the roller which imparts to the paper the forced motion around *B*. The pencil oscillates around a fixed point no



TOOTH-CURVE DRAWING INSTRUMENT.

shown in the drawing. The position of the point A is determined by the distance of the point O (Fig. 2) from A, equal to $R+\epsilon$, and by the length a. The amplitude of oscillation of the drawing pencil is 50 mm., and the radius ρ of the generatrix is 100 m. The trace of the tooth outline is found in the following manner: Given R=pitch circle radius of wheel A; R_1=pitch circle radius of wheel B; ρ=radius of curvature of flank of tooth of A; ϵ=distance of centre O of this flank from pitch circle A; a=height of tooth above pitch circle; i = height of tooth inside pitch circle; t=pitch; θ=thickness of tooth of wheel B; θ_1=thickness of tooth of wheel A; These dimensions, reduced in the ratio of $\frac{\rho}{100}$, are respectively designated R^{100}, $R_1{}^{100}$, &c. We have now to adjust the pivot A, making $AC=R^{100}$ and $BC=R_1{}^{100}$, in such a manner that the roller C (Fig. 3) describes a circumference distant by a^{100} from the extreme position E of the oscillating lead pencil, and by ϵ^{100} from the centre of rotation of the generatrix. Having fixed the paper at B, we oscillate the pencil while swinging the beam compass with the paper that turns about B and is pressed against the drawing-board by the roller C. The pencil will describe the envelope of the tooth of wheel B conjugate to the assumed tooth for wheel A. The construction of tooth profiles for sets of wheels may also be done with the instrument.—*Amer. Machst.*, *Vol. XXVI*, *No. 47*, pp. 1624–1626.